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THE INSTITUTE OF PETROLEUM

AN Ordinary General Meeting of the Institute of Petroleum was held at 61 New Cavendish Street, London, W.1, on 4 December 1957, the Chair being taken by J. S. Parker, a member of Council.

The General Secretary read the minutes of the previous meeting, which were confirmed and signed as a correct record. He also announced the names of members elected since the previous meeting.

The Chairman, introducing the author of the paper to be presented, said:

Mr Downs is a B.Sc. in Engineering, with First Class Honours; he joined the experimental laboratory of Ricardo and Company and has been engaged in investigation of gasoline engine performance and various aspects of the applications of fuel and lubricating oils to the internal combustion engine. Since 1947, he has been in charge of a department, having special responsibility for these things, and in 1957 was appointed a director of the company. Mr Downs is also the author of a number of papers on engine combustion and lubrication problems.

The following paper was then presented in summary.

RECENT ENGINE DEVELOPMENTS IN RELATION TO FUELS AND LUBRICANTS *

By D. DOWNS † (Associate Fellow)

INTRODUCTION

THE internal combustion engine as a prime mover has been in use now for some 80 years, and 1959 will see the hundredth anniversary of the sinking of the first oil well in the U.S.A. It might have been expected, therefore, that the pattern of engine development, on the one hand, and of petroleum usage, on the other, would have settled down by now into a fairly rigid mould. This is very far from being the case, however, and, stimulated to some extent by the increase in research activity during and after the second world war, the production and user pattern is in as fluid a state now as it has ever been. A number of the current developments in engine design are, in fact, such as to disturb the existing pattern of supply distribution between the various types of prime mover and the various grades of petroleum fuel. These changes pose problems for both the oil supplier and the user which will demand their combined efforts and goodwill to overcome. This would seem, therefore, to be a useful time to attempt a review of these developments with the object of divining the future trends. The present paper makes no attempt to be exhaustive, but attention has been concentrated on those fields where change is more evident and where the author has greater personal knowledge of developments.

THE SPARK IGNITION ENGINE

Compression Ratio

In the spark ignition engine field there has over a long period been a steady rise in compression ratio and a corresponding increase in the O.N. of the gasoline used. Engine efficiency increases with compression ratio, but as Fig 1 shows, it is a law of diminishing returns. The top curve shows the relationship between the theoretical air standard efficiency and compression ratio. The next curve shows the indicated thermal efficiencies actually obtained on the Ricardo E.6 variable compression engine over the same range of compression ratio. It can be seen that the relative efficiency of actual/theoretical remains fairly constant over the compression ratio range.

The actual brake thermal efficiencies will be somewhat lower than the indicated efficiencies due to friction, and this, like the efficiency, will increase with compression ratio. It is perhaps unfair to assess the effect of compression ratio on a variable compression engine because this type of unit, built to withstand the high pressures associated with the high compression ratios, will penalize the relative performance at low compression ratios due to the high friction of its necessarily large bearing areas.[1] For this reason the brake thermal efficiency curve of Fig 1 has been derived from the indicated curve using, not the friction figures obtained on the E.6 engine, but those derived by Bastow [2] from multi-cylinder engines of various compression ratios. It can be seen that the mechanical efficiency decreases only slightly, from 90 per cent at a C.R. of 5 to 88 per cent at a C.R. of 16. The improvement in brake thermal efficiency on increasing the compression ratio from 5 to 15 is 40 per cent, a figure in good agreement with that derived by Withers and Eatwell.[3]

All these figures are for maximum efficiency at full throttle, but it must be remembered that the petrol

* MS received 15 October 1957.

† Ricardo & Co., Engineers (1927) Ltd.

road vehicle spends a large proportion of its time at part throttle, where the mechanical losses loom relatively larger and where their increase due to compression ratio is more important. This is illustrated by the lowest curve of Fig 1, which shows the brake thermal efficiency curve for $\frac{1}{4}$ load flattening off more markedly at a compression ratio of 15 than the corresponding full throttle curve.

Fig 2 shows how compression ratio has increased in British vehicles [4] compared with those made in the U.S.A.[5] It can be seen that in 1954 the average compression ratios of British and American cars were very similar. Since then, however, compression ratio has risen much faster in America than in the U.K. and the average compression ratio of 1957 vehicles was 8·75 in the U.S.A. as compared with only 7·65

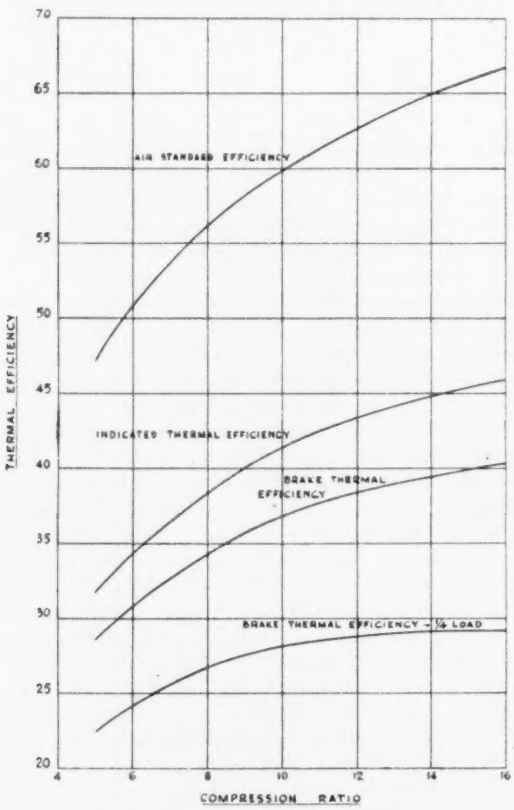

Fig 1

EFFECT OF COMPRESSION RATIO ON THE THERMAL EFFICIENCY OF A SPARK IGNITION ENGINE

in Britain. The O.N. of premium fuel in Britain is very similar to that in the U.S.A., although British regular grade fuel is some 8 O.N. lower. This latter fact may tend to retard the increase in compression ratio in the U.K., as some of the more popular cars are designed to use regular rather than premium grade fuel. Apart from this, there seems no reason why compression ratios in Britain should not rise as high or higher than those in the U.S.A. For one thing, British cylinder sizes are smaller and engine

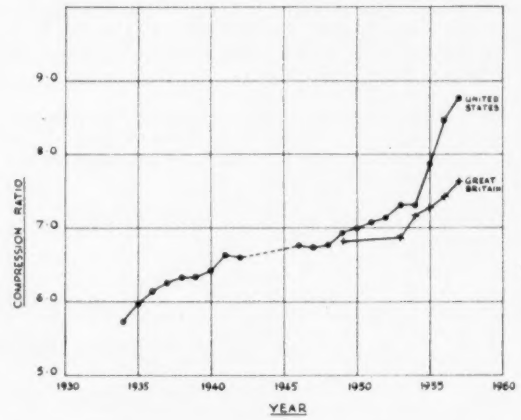

Fig 2

INCREASE IN COMPRESSION RATIO OF PETROL ENGINES IN GREAT BRITAIN AND THE U.S.A. IN RECENT YEARS

speeds are higher, both factors tending to reduce octane requirement. Against this must be set, first, the widespread use of automatic transmissions in America, which restrict full throttle low speed operation, and secondly, the generally lower load factor at which their engines operate. This latter consideration, besides reducing octane requirement, enables the American designers to use lighter scantlings for a given compression ratio. With the increasing use of common cylinder and crankcase units for both petrol and diesel versions of essentially the same vehicle engine, however, there seems no reason why the petrol version need be restricted on mechanical grounds from using the highest economical compression ratio.

Looking ahead, therefore, it would seem reasonable to expect that compression ratios in the U.K. would rise to the present American figure of 8·75 and beyond to about 10, which there is a good deal of evidence to suggest is the figure which it will be possible to use with the 100 O.N. fuels now being marketed in the U.K. and the U.S.A.

Fig 1 suggests that a further gain in brake thermal efficiency of 8·5 per cent could be obtained by raising the compression ratio from 10 to 15, after which point the gains to be obtained seem very small, particularly if part load running is also brought into consideration. The O.N. required to satisfy an engine operating at a compression ratio of 15 is a matter of conjecture. Extrapolating figures obtained on the E.6 engine gives a value of 110 for a compression ratio of 12·5, and this agrees with other estimates.[6] The gain in brake thermal efficiency on increasing the compression ratio

from 10 to 12·5 is 5 per cent, and so to be economic the price of the fuel required to satisfy the higher compression ratio must not be more than 5 per cent greater than that for the lower. Taking as a basis

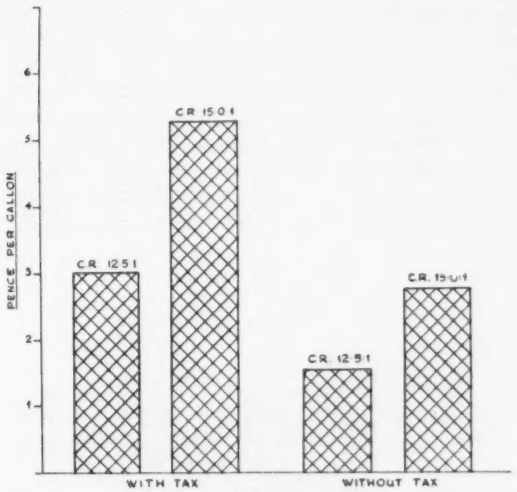

FIG 3

THE EFFECT OF COMPRESSION RATIO ON THE ECONOMIC PRICE INCREMENT FOR GASOLINE ABOVE 100 O.N.

a 100 O.N. fuel retailed in the U.K. at 5*s* 1½*d* per gallon as satisfying a 10 : 1 C.R. engine, Fig 3 shows the maximum increased price which it would be economic for the customer to pay for fuel fully satisfying a 12·5 : 1 and a 15 : 1 compression ratio car. Figures are also given for the same fuel, assuming that it was marketed without the present tax of 2*s* 6*d* per gallon. It can be seen that were it not for the high rate of motor tax, the economic price increment for the higher octane fuels would be halved. High tax, therefore, puts a premium on fuel quality, but it is not yet clear whether the oil companies can produce a fuel of the required O.N. at a price which it would be economic for the customer to pay.

So far there is little sign of a special combustion chamber being developed for high compression ratio engines. Most of the chambers in use are adaptations of standard forms with the clearance volume suitably reduced. The side valve engine becomes very rough and noisy above a compression ratio of about 7, and is also restricted in its breathing at high speed. This form has, therefore, virtually disappeared. Most combustion chamber shapes are variants of the inclined " bath tub " or " F " head, and these perform satisfactorily provided care is taken to avoid masking the valves and thus impairing the volumetric efficiency. The hemispherical head, long thought the ideal at lower compression ratios, loses much of its attraction at higher compression ratios, where to reduce the clearance volume it is necessary to dome the piston, thus giving a rather unattractive " orange peel " shape, which is also poor from the octane requirement point of view. The special combustion chamber used by the Rover Company (Fig 4) is an example of a design which seems to suffer least distortion of combustion chamber shape as the compression ratio is raised. A form of combustion chamber having an ante-chamber, as on some types of diesel, has sometimes been suggested as the best means of disposing of the small clearance volume associated with high compression ratios.[7] Combined with petrol injection, this would also hold out the possibility of improving efficiency by means of charge stratification at light load.

The high compression engine will undoubtedly have its special fuel and lubricating oil problems. Such an engine will be sensitive to combustion chamber deposits, and the higher temperatures both of the compressed mixture and of certain sensitive hot spots

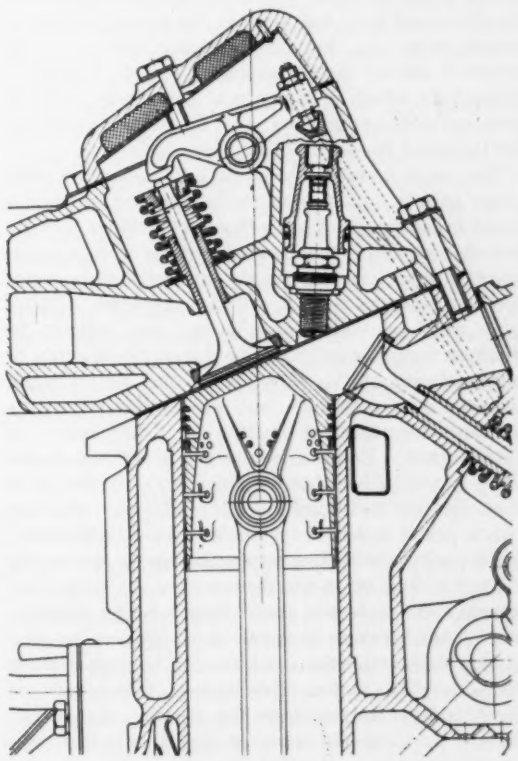

FIG 4

ROVER 105 COMBUSTION CHAMBER
C.R. : 8·5

will enhance the importance of the pre-ignition tendency of the fuel. The higher octane fuels will undoubtedly require high T.E.L. contents for their economic achievement. Although British premium grade fuel has about the same O.N. as that sold in

the U.S.A., the lead content is only about one-half that of the American. As O.N. increase, the lead content used in the U.K. may be expected to approach the American figures. American experience shows that engines can be developed to operate happily on highly leaded fuel, but improvements in valve material and valve design will be required, and one wishes that there were more signs of British engine manufacturers preparing for this eventuality.

Petrol Injection

Few topics concerned with the automobile engine have been the subject of as much discussion in post-war years as petrol injection. Despite the development of some promising systems, however, it has not found general acceptance, except for certain specialized applications. The fact is that the carburettor has been developed to give a reasonably satisfactory performance and, although petrol injection can be shown to do a better job in certain circumstances, the extra cost does not warrant the change except in special cases. In the aircraft engine field petrol injection is already well established, and the important advantages which it gives from the safety point of view are well appreciated, and for that reason it is not proposed to elaborate on them here.

The term "petrol injection" is sometimes used rather loosely, but it is generally intended to cover a timed injection either into the inlet port or directly into the cylinder. Some other systems of continuous injection might more properly be described as injection carburettors. With a timed injection system, some 5–10 per cent more power can usually be obtained from the same engine due to the absence or increased size of the choke tube, the use of larger diameter inlet manifolds, and the elimination of the necessity for manifold heating by "hot spot" or water jacket. Petrol injection *per se* will not necessarily give a higher thermal efficiency. In fact, it is often difficult to obtain the same thermal efficiency with a petrol injection system as with a carburettor, due in part to the shorter time available for the mixing of the fuel and air in the former case. A better fuel economy can, however, result from a better distribution of the mixture between the cylinders by permitting an overall weaker mixture to be used without misfiring. This better distribution of fuel between the cylinders, coupled with the absence of manifold heating and, in the case of direct injection, the possibility of using the fuel spray to cool overheated parts of the combustion chamber, in general permits a higher compression ratio to be used with petrol injection for a given quality fuel. This is a further source of increased economy. With a petrol injection system also, control of fuelling during the over-run is facilitated, which not only improves economy, but also helps in the reduction of atmospheric pollution. Petrol injection, and in particular direct cylinder injection, gives improved flexibility in operation, due to the instant response of the fuel system to the demands made on it by the engine.

The two-stroke engine shows to particular advantage with petrol injection because it eliminates the loss of fuel during the scavenging process, thus not only improving economy, but also enabling the power output to be raised by increasing the scavenge period. The German Guttbrod and Goliath engines [8] are the outstanding examples of this line of development. Special difficulties in the application of petrol injection to two-stroke engines are the high speed of operation and fuel control, the induction air flow giving little indication of the engine's fuel requirements.

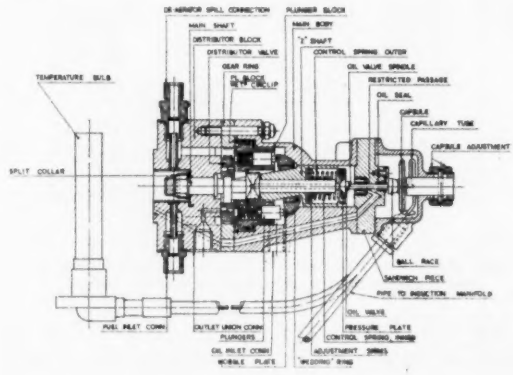

FIG 5

THE SU FUEL INJECTION PUMP

The system used on the Mercedes 300 SL [9] is probably the outstanding example of direct cylinder injection on a four-stroke unit. A Bosch jerk-pump injection system is used, as on the diesel engine, with a mixture regulator controlled by a pressure-sensing unit built into the intake pipe of the engine in place of the carburettor. A conspicuous feature of the 300 SL with petrol injection is its remarkable flexibility over the speed and load range.

A direct injection system really demands a combustion chamber specially designed for it, as it was on the Mercedes, and this rather restricts its application. Most of the advantages of petrol injection can, however, be obtained by port injection, and this is more flexible in its application. Additionally, the nozzle, not being exposed to the effects of temperature, pressure, and carbon formation in the combustion chamber, should have a longer life. Two port injection systems which have been developed in the U.K. are the SU [10] and the Lucas.[11] The former uses a multi-plunger swash plate pump of a type first deloped as a fuel feed pump for aircraft engines during the war. It was later adapted for timed port injection on the Blackburn Bombardier aircraft engine and is now used on military vehicles in Britain and the U.S.A. (Fig 5).

It is also being developed for use on automobiles. The control of fuel delivery is of the speed-density type, the pump being engine-driven and its output at a given speed being controlled by pressure- and temperature-sensing devices placed in the inlet manifold. This type of system assumes that the air consumption of the engine is proportional to the product of cylinder volume and the pressure and temperature in the intake port. It has the virtue of simplicity, but is only approximately correct, particularly under conditions of wide valve overlap.

The Lucas system (Fig 6) uses a feed pump to supply fuel to an engine-driven distributor which meters the flow and passes it in timed sequence to each poppet injection nozzle in turn. Control of the

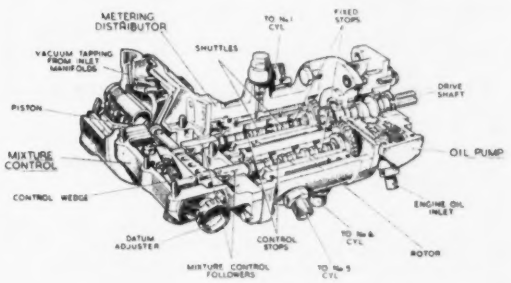

FIG 6

LUCAS PETROL INJECTION: THE METERING UNIT FOR THE JAGUAR SIX-CYLINDER ENGINE

mixture supplied by the distributor is by a venturi in the induction system. This injection system, which has been used in the U.K. on the racing Jaguar engines, is being exploited in the U.S.A. by the Holley Carburettor Co.

The only American car currently offering petrol injection as an optional extra is the Chevrolet,[12] but this system is really an injection carburettor spraying through injectors into the individual inlet ports. American Bosch are developing a system incorporating a single plunger pump and distributor.[13] Bendix are developing a common rail system with a non-metering fuel pump supplying fuel at 20 p.s.i. to individual poppet nozzles which are electromagnetically operated, the quantity of fuel injected being electronically controlled on a speed-density basis.[14]

One potential advantage of petrol injection is that it holds out the possibility of enabling one to work with quality control through charge stratification, and hence achieve a better thermal efficiency under light load conditions. Attempts to achieve this have not, however, been very successful except on a sleeve valve engine, where the regularity of the air movement enables the path of the mixture to be predicted within limits. With a poppet valve engine, the indiscriminate air movement within the combustion chamber tends to spoil any pre-set mixture distribution pattern. An attempt is sometimes made to get over this difficulty by using an ante-chamber, into which the fuel is injected, to contain the rich mixture around the sparking plug required for light load running. With this arrangement it is usually possible to obtain good light load running and a good performance at full load, but there is an awkward gap in the mid-load range where combustion is poor, and which it has so far been found impossible to bridge satisfactorily.

Even if charge stratification were possible, it would be difficult to obtain a mixture range loop as good as that normally obtained on the diesel engine, with the disadvantage in the former case of having to use both injection and ignition equipment. On the other hand, the maximum pressures, and particularly the maximum rate of pressure rise, would probably be lower on the petrol injection engine than on the diesel unit, and this would give a quieter and a lighter engine. There might be more of a case for petrol injection if the normal combustion chamber development at high compression ratios moved towards an ante-chamber design and this was also found suitable for petrol injection and charge stratification.

The effect of the introduction of petrol injection on fuel volatility requirements is difficult to establish. From the point of view of distribution between the cylinders, and because of quicker engine warm-up with petrol injection, it should be possible to reduce the volatility of the fuel, but there is evidence that it would not be possible to go far in that direction without sacrifice of combustion efficiency. All the petrol injection systems are pressurized, and this has led some manufacturers to claim that fuel volatility can be increased without risking vapour lock. Only extended service trials would show whether this was in fact the case.

THE DIESEL ENGINE

General

Two important developments of the diesel engine in recent years have been the extension of the field of utility of this type of unit to the smaller sizes, which were once thought to be the exclusive domain of the petrol engine; and, on the somewhat larger sizes, the increasing use of supercharge, both by positive-displacement blowers and with turboblowers. Before proceeding to discuss these two developments in more detail, it might be of interest to consider the fundamental difference in the performance characteristics of petrol and diesel engines. For this purpose, some figures obtained on the Ricardo E.6 engine with a petrol cylinder head at compression ratios varying from 5 to 13 and with a diesel head of fixed compression ratio and having a Comet Mk. V combustion chamber, will serve as an illustration. Of course the friction losses of this type of engine are high and the

brake performances correspondingly low, but both cylinder heads are penalized equally.

It can be seen from Fig 7 that the power output of

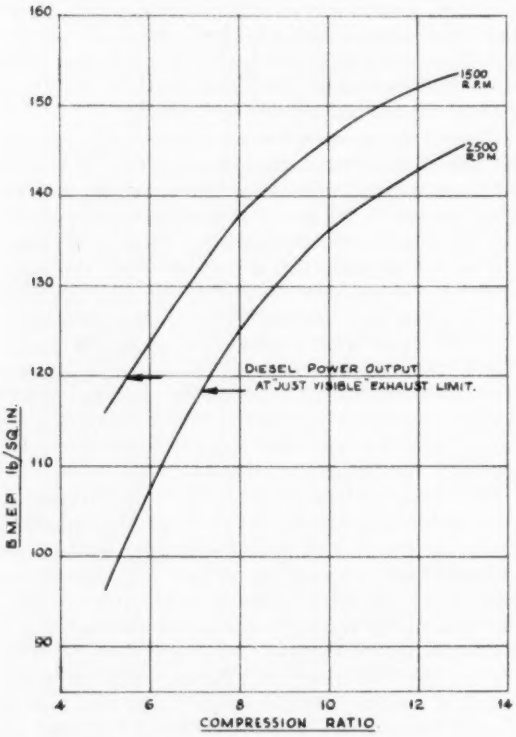

FIG 7

EFFECT OF COMPRESSION RATIO ON THE POWER OUTPUT OF THE PETROL ENGINE, SHOWING THE COMPARISON WITH THE DIESEL ENGINE

the diesel engine at a " just visible " exhaust smoke limit matches that of the petrol engine at a compression ratio of 5·5 at 1500 rev/min and 7·2 at 2500 rev/min. No heat was added to the intake air on the petrol engine and the volumetric efficiencies of the two versions were in very close agreement over the speed range. In one sense the comparison is unfair to the petrol engine as both versions were restricted to the same valve arrangement and cam timing, whereas in the practical case the petrol engine has rather more freedom in these respects, and in general can have a wider valve overlap and larger valves for a given cylinder volume. On the other hand, the Comet Mk. V combustion system used on the diesel version has a very high air utilization, over 95 per cent at 1500–2500 rev/min, and this raises its power output compared with what would be obtained from, say, a larger direct injection unit. The general conclusion, however, supports the general experience that the power output of the diesel engine compares unfavourably with the best performance possible on the petrol engine.

When we come to consider economy, however, the position is very different. Fig 8 shows the comparison of the full throttle brake specific fuel consumptions of the two units on a weight basis. On both sets of curves the figure for the fuel consumption at the " just visible " power output of the diesel engine has been superimposed on the maximum fuel consumption curve of the petrol engine, and the minimum fuel consumption of the diesel engine has been superimposed on the corresponding curve for the petrol engine.

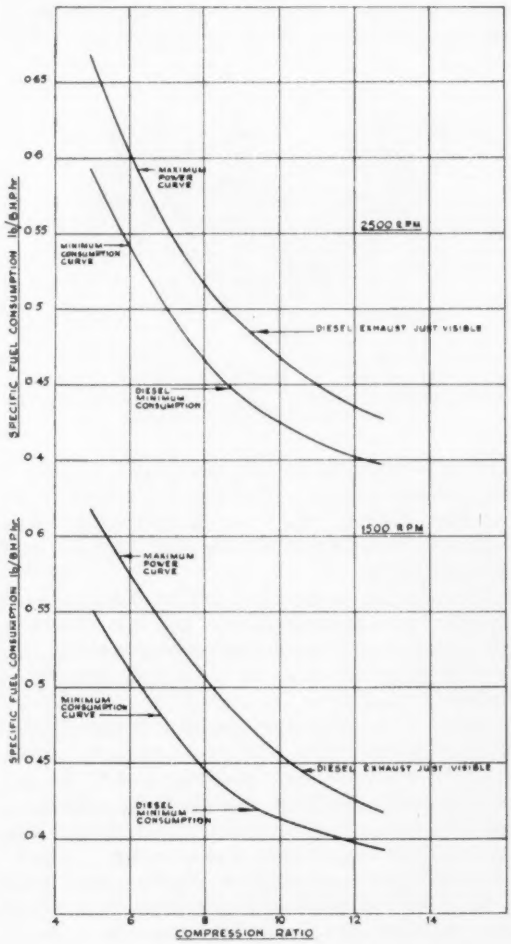

FIG 8

EFFECT OF COMPRESSION RATIO ON THE SPECIFIC FUEL CONSUMPTION OF THE PETROL ENGINE, SHOWING THE COMPARISON WITH THE DIESEL ENGINE—WEIGHT BASIS

It can be seen that the consumptions are approximately equal for both conditions at a compression ratio of 10 at 1500 rev/min and a compression ratio

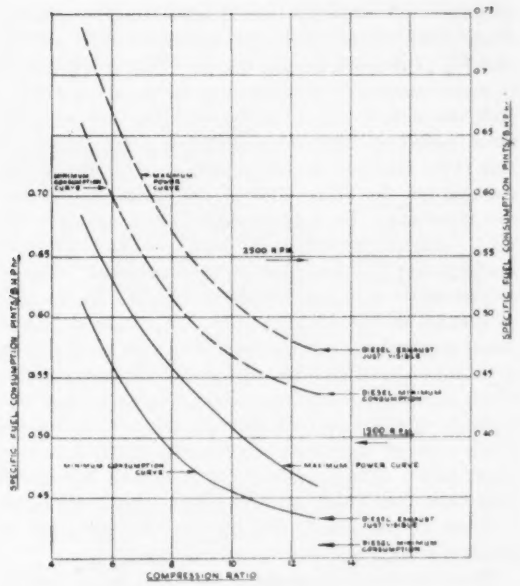

FIG 9

EFFECT OF COMPRESSION RATIO ON THE SPECIFIC FUEL CONSUMPTION OF THE PETROL ENGINE SHOWING THE COMPARISON WITH THE DIESEL ENGINE—VOLUME BASIS

of 9 at 2500 rev/min. It is not suggested that these figures of limiting compression ratio have any absolute significance. On large cylinder sizes, for instance, where a direct injection combustion chamber could be used, it is probable that some 5 per cent improvement in economy could be obtained on the diesel engine with some sacrifice in maximum power. The curves do, however, illustrate trends and show that with the compression ratios now being attained with modern petrol engines, 10 and above, the full load specific fuel consumptions of the two systems are very similar.

So far it is specific fuel consumptions on a weight basis which have been referred to, and this is the basis used by the engineer when considering relative efficiencies. The automobile customer, on the other hand, buys his fuel on a volume basis, and here the picture is even more favourable for the diesel engine (Fig 9). At 2500 rev/min the petrol engine has just about equalled the economy of the diesel engine at a compression ratio of 13. At 1500 rev/min it is evident from the form of the curves that equality will not be achieved even at a compression ratio of 15.

Fig 10 illustrates the further big advantage of the diesel engine, its good economy at light load. On this figure have been drawn a series of throttle loops on the petrol engine at 2500 rev/min and a compression ratio of 8 and compared with this a load range curve for the diesel engine at the same speed. It can be seen that, at full load, the specific fuel consumptions are very similar, as would be expected from Fig 8. At light loads, however, the diesel curve lies wholly below that for the petrol engine; at half load showing a 30 per cent better economy than the weakest mixture for maximum power on the throttled loop.

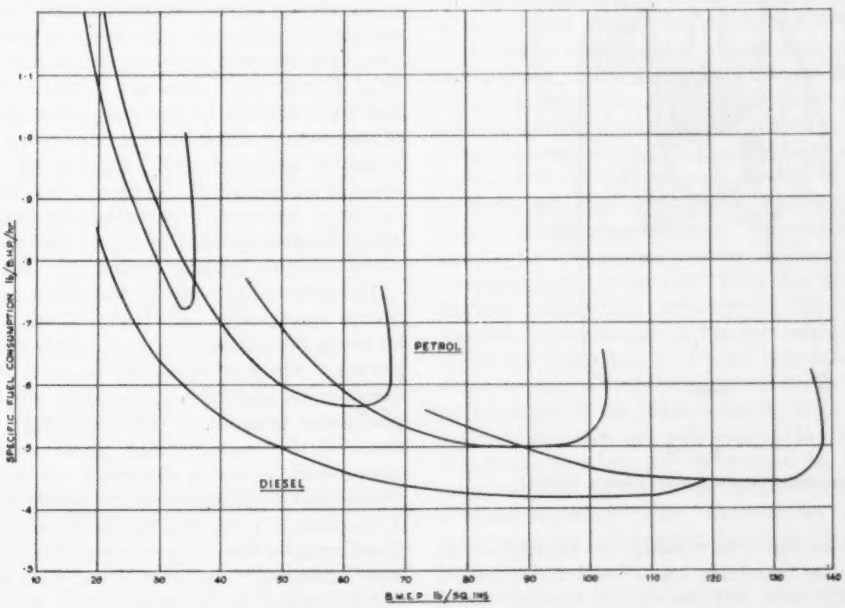

FIG 10

COMPARISON OF THE PETROL AND DIESEL ENGINES OVER THE LOAD RANGE

The continuing development of the petrol engine utilizing higher compression ratios will bring the full power specific fuel consumptions of the petrol and diesel units very close, and it may be that for applications where high-speed, high-power outputs are required continuously, the petrol engine will gain some ground from the diesel engine on account of the greater comfort and absence of noise of the smoother prime mover. In this connexion, a possible portent was the Rolls Royce-engined Harrington coach shown at the last Commercial Motor Show.[15]

The Small Diesel Engine

For intermittent stop–start duty, such as the city bus service, the diesel engine has long been supreme,

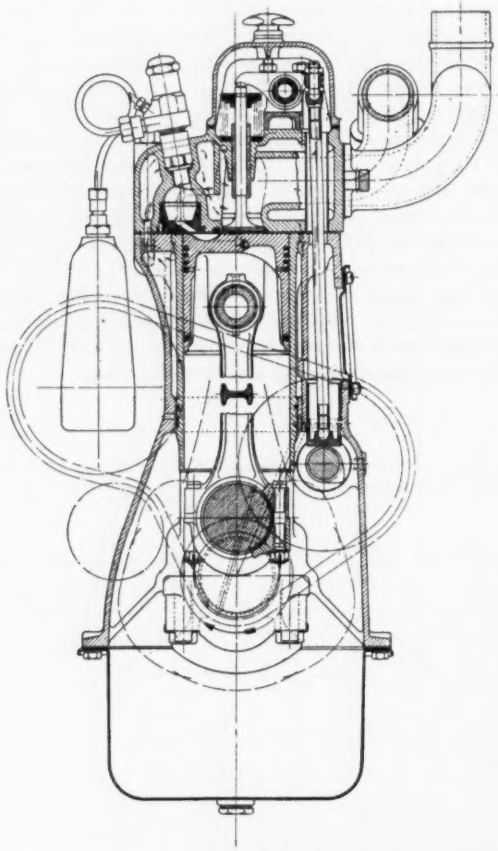

FIG 11
CROSS-SECTION OF SMALL DIESEL ENGINE

and there is no sign of any change in this application. Equally intermittent duties are demanded of taxi-cabs, small delivery vans, and agricultural tractors, and it was only the absence of a suitable design of small diesel engine which had until recently prevented their adoption. A large number of small diesel engines of about 2 litres in four cylinders are now on the market, and Fig 11 shows a typical design. These engines are in many cases able to operate up to speeds of 4000 or 4500 rev/min, which is not far from the normal maximum speed of 5000 rev/min for a petrol engine of this type, and give a torque curve which in the best cases is not far below that of the petrol engine they are replacing. In a particularly good example 130 p.s.i. maximum B.M.E.P. was obtained at 2500 rev/min and 31·7 h.p/litre at 4000 rev/min. Careful comparative tests on London taxi-cabs shows that in normal service 34 m.p.g. can be obtained from a diesel-engined vehicle as compared with 18 m.p.g. on a petrol-engined vehicle of equivalent performance. With such a fuel consumption differential, the high mileage of the taxi-cabs enables them to offset the extra cost of the diesel engine (£80–£100) in a very short time. The application is in fact so favourable that only four years after the first large-scale introduction of the diesel cab, nearly 100 per cent of London's taxis are now diesel-operated.

These same diesel engines are now being used in small vans of all types, and in time there should be as spectacular an increase in this application as there has been in the taxi-cab and the agricultural tractor. For fiscal reasons the agricultural tractor used to operate with the very inefficient kerosine engine. Diesel fuel, unlike gasoline, enjoys the same tax exemption as kerosine when used for agricultural purposes, with the result that production of these small tractors is now over 90 per cent diesel. The small vehicle diesels are increasingly being used for industrial applications, together with small single and twin cylinder units specially designed for the purpose (Figs 12 and 13). Although in the very small sizes first cost tends to be more important than economy of operation, a single cylinder air-cooled diesel engine is already marketed with a capacity of only 350 cc.[16] One big advantage of the diesel over the petrol unit for small industrial applications is that it is more easily weather-proofed owing to the absence of the water-sensitive ignition system.

The greater first cost of the diesel engine and its noisier operation at light load and idling will restrict its use in the private car, particularly where mileages are low. The cost differential between the petrol and diesel units might be expected to diminish as the production numbers of the latter begin to approach those of the more popular petrol engine. As the compression ratios of passenger car engines are increased and scantlings have to be stiffened accordingly, it becomes a more practicable proposition to fit a diesel cylinder head to a common diesel/petrol crankcase. Bedford [17] and Hercules [18] have already done this in various engine sizes and, in time, this could be an important factor in cheapening the small diesel engine. The noise question is one to which much

attention is being given today, and the improvements already made in engine mounting and bonnet sound insulation have done much to minimize the trouble.

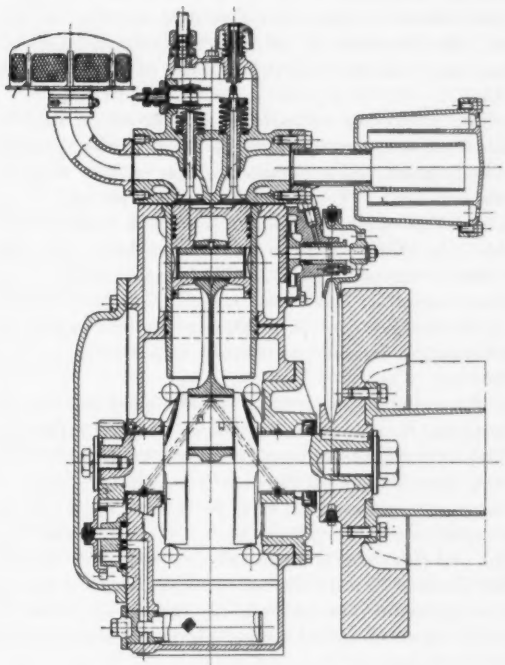

FIG 12

LONGITUDINAL AND CROSS-SECTIONAL ARRANGEMENT OF SINGLE-CYLINDER COMET MK. III—76 × 76

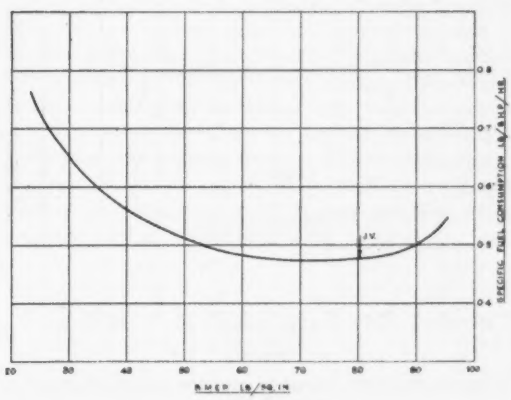

FIG 13

LOAD RANGE CURVE AT 2000 REV/MIN SINGLE CYLINDER INDUSTRIAL DIESEL ENGINE

On the whole it would seem that the first automobile application of the diesel engine will be to the medium-sized car used by firms in large fleets for their representatives. The high mileage of these vehicles, combined with the cost-conscious conditions under which they are operated, and the facilities for servicing which could exist, make this a favourable application. A car of this type, which has been operated by my company for the last 3½ years and which has accumulated 42,000 miles, has averaged over its whole life some 40 m.p.g. This compares with the performance of the same type of vehicle with a smaller capacity petrol engine of similar power output, where a fuel consumption of only some 27 m.p.g. would be expected.

The Supercharged Diesel Engine

The intermittent duty road vehicle engine is one application where the diesel engine shows to particular advantage. At the other extreme is the heavy duty application where high power and high torque are required, and here the readiness with which it lends itself to supercharge again renders the diesel engine supreme. Supercharging the diesel engine can readily give a 40–50 per cent increase in power output with corresponding reductions in bulk/weight and power/weight ratios, and, in most cases, of cost. This is a development which favours the diesel engine at the expense of the petrol, because in the former case supercharging makes no extra demands on the quality of the fuel. In fact, supercharging in most cases very considerably widens the range of fuels which can be burnt in the diesel engine, as well as giving quieter and smoother combustion. Two types of blower are in general use; the mechanically-driven positive-displacement type and the turboblower, and each seems to have its particular field of application. The turboblower shows to particular advantage in the larger engines under steady speed, fairly steady load conditions typical of industrial and marine applications and of the diesel locomotive with electric drive. Although some types of turboblower are being developed for road vehicle engines [19, 20] and will probably have their application for high-speed, inter-city bus or truck operation, the positive displacement blower [21] seems a more logical solution for city service and for the important earth-moving equipment market. With this form of supercharger, unlike the turboblower, the air delivery pressure is maintained at the low engine speeds required for these types of road vehicles. A turboblown engine, on the other hand, tends to have its peak torque at an inconveniently high speed, the engine often does not pull well at low speeds, and vehicle acceleration may be delayed by the time required to speed up the turbine motor. Some, if not all, of these objections may, however, be met by the use of a torque converter if the additional cost can be faced.

Fig 14 shows the performance characteristics of an engine fitted with the Wellworthy–Ricardo positive displacement blower.[21] With the standard valve

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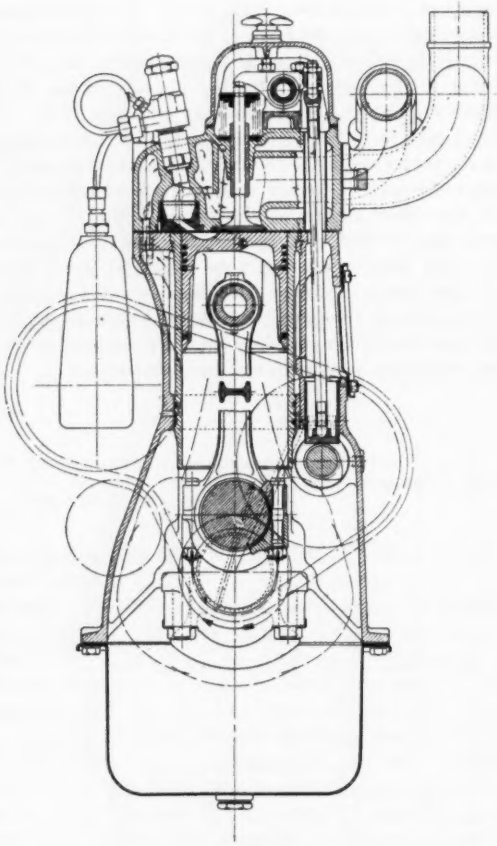

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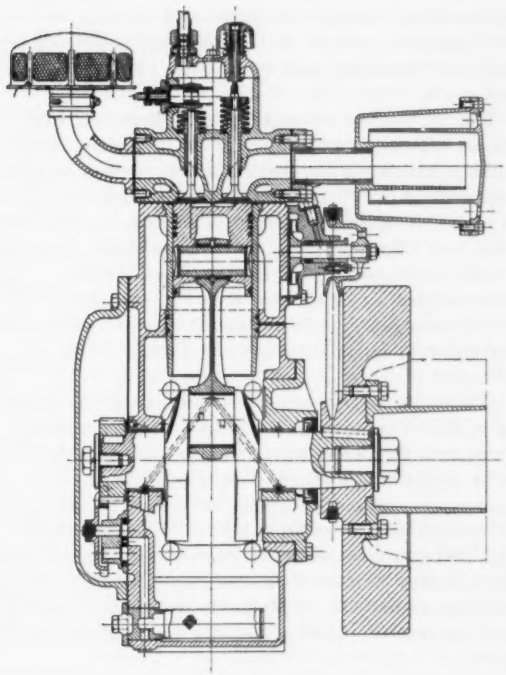

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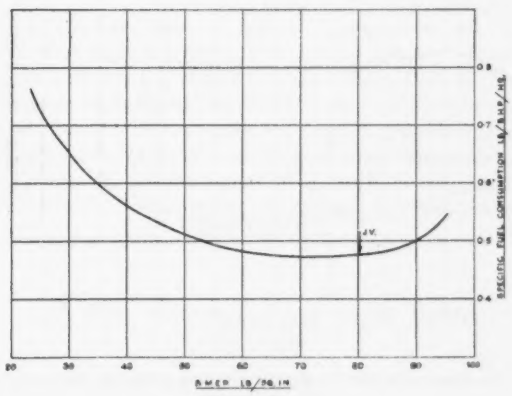

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Fig 14 shows the performance characteristics of an engine fitted with the Wellworthy–Ricardo positive displacement blower.[21] With the standard valve

porting the blower, with open delivery, gives a substantially constant volumetric efficiency throughout its whole speed range, but by varying the form, or the time of closing of the inlet port, the volumetric

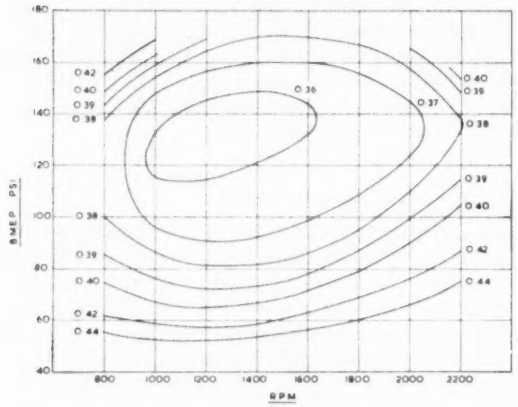

FIG 14

PERFORMANCE CHART FOR SIX-CYLINDER 6-LITRE D.I. ENGINE SUPERCHARGED BY WELLWORTHY–RICARDO SUPERCHARGER

Mean boost pressure 19″ hg
Fuel consumption in pints/b.h.p/hr

efficiency curve can, at will, be given either a rising or drooping characteristic with increase of speed. This type of blower can be used to make a small engine do the work of a larger one—with this size of blower a 5-litre engine could give the same performance as a normally aspirated 8-litre engine, but with a considerable saving of bulk, weight, and, in some cases, cost. On the other hand, with the improvement in the road conditions and a gradual lifting of the speed limit of heavy vehicles, the demand for power for these units will increase. The general level of power output for coaches in the U.K. is 125 h.p. The direction in which these might go in the future is shown by the new 39-seater Greyhound bus in the U.S.A. with a 211-h.p. G.M. engine. To fit an engine of this power output means either designing a new unit of larger size or utilizing an existing engine and getting the increased power output by supercharging. As well as giving a smaller bulk, lighter weight unit, this latter solution could give a much cheaper one by enabling production to standardize on the one size of engine and cover the power range required with supercharge.

The Fuel Requirements of the High-speed Diesel Engine

So far as the fuel requirements of the high-speed diesel engine are concerned, the principal limitations, apart from cold starting, are misfiring at high speed and light load and noise at low speed. The diesel fuel currently marketed in the U.K. for road vehicles has a C.N. of 50–55. Improvements in starting can be obtained by increase in C.N. up to 70.[22] It would be difficult to achieve this C.N. without the use of ignition accelerators, which in any case do not give a commensurate improvement in cold starting, so that any shortcomings of the engine combustion system have to be made good by the use of extra starting aids.

The possibility of marketing a premium fuel for diesel engines, on the lines of the premium gasoline already marketed for spark-ignition engines, is sometimes mooted. There might be a case for such a fuel if high C.N. gave a big improvement in noise at light load and idling, because there is no doubt that this would enormously increase the acceptability of the diesel engine for small road vehicles. In the author's experience, however, it is difficult to achieve a marked reduction in combustion noise by C.N. increases above 55.

The diesel engine is nevertheless fairly omnivorous in its fuel requirements, the use of a high compression ratio or a moderate degree of supercharging enabling even petrol to be burned satisfactorily if, with, in some cases, an increase in noise level. Fig 15 shows comparative performance figures for diesel fuel and M.T. 80 petrol on a single-cylinder engine fitted with the Ricardo Comet V combustion chamber having a compression ratio of 20·6. It can be seen that the performance on petrol is very similar to that obtained

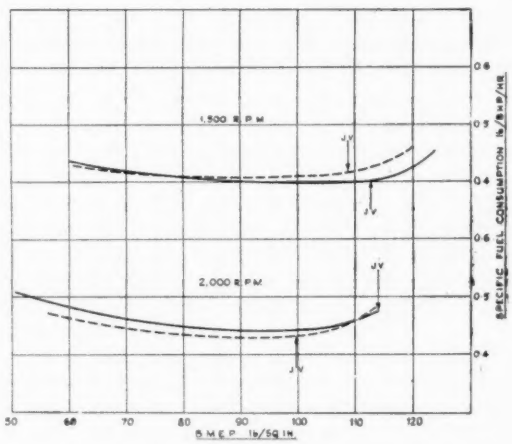

FIG 15

COMPARATIVE PERFORMANCE FIGURES FOR GAS OIL AND M.T. 80 PETROL

M.T. 80 ———— Gas oil — — — — —

with diesel fuel, with, as might be expected, some fall-off at higher speeds and lighter loads.

The Marine Diesel Engine

In recent years the power output of the marine diesel engine has been steadily increased, mainly by

turboblowing, in an effort to extend the range of size of ship to which the diesel engine can be applied. In order to compete economically with the steam turbine, it has become practically mandatory for the marine diesel engine to be capable of burning residual fuel, if not 3500 sec. Bunker " C," then the intermediate 1500 sec. blend. The principal drawbacks to the use of the heavier fuel are the increased cost of processing the fuel on board ship and the increased cost of engine maintenance, including earlier replacement of the cylinder liners. Despite these disadvantages, such is the price differential between residual fuel and marine diesel fuel, that it has been estimated that the use of the former can involve a net saving of £2000/year/cylinder in operating costs. Improvements in the treatment of the fuel and in the materials used for cylinder liners and piston rings have reduced the wear when running on residual fuels. Perhaps the most spectacular improvements have come from the use of the new cylinder lubricants of high additive content. To get sufficient quantity of low cost additive into the fuel, it is necessary to get away from oil-soluble additives and to go to an emulsion-type oil [23] or to a dispersant type.[24]

With the emulsion type of oil, van der Zijden and Kelly [23] have found that the wear with residual fuel can be reduced to the same figure as is obtained with a marine diesel fuel using normal lubricant. Even with this big improvement, however, there is a 10 : 1 ratio between the wear rates of the best and the worst engines. Further big improvements in cylinder life should result from a better understanding of the mechanism of wear. One simple fact, which at the moment is imperfectly understood, is the proportion of wear with residual fuel due to corrosion from the high sulphur content and the proportion due to abrasion from the ash and from the harder carbonaceous particles. Better understanding of the fundamentals of this problem should help us in finding the direction in which to go for further improvement. Apart from the pure economics of liner replacement, shortage of skilled engine room staff is making it difficult for ship owners to operate a plant which demands a lot of skilled attention.

THE GAS TURBINE AND FREE PISTON ENGINE

Industrial Applications

The gas turbine, apart from one or two isolated examples of designs for industrial and locomotive purposes, really got started during the war for aircraft applications, and its development since has been such that last year for the first time the world demand for aviation turbine fuel exceeded that for aviation gasoline.[25]

Since the war there has been a tremendous effort, backed by considerable research facilities, to expand the field of application of the gas turbine to industrial uses. The early excessive enthusiasm for the new form of prime mover has now given way to a more sober assessment of its position. The big advantages of the gas turbine over other power plants are low first cost, high power/weight and power/bulk ratios, and independence of water supply. Its disadvantage is its poor thermal efficiency compared with the diesel engine and this substantially confines its use for continuous operation to applications such as oilfield power generation, where natural gas provides a cheap fuel, and water for cooling a diesel engine may be scarce. There are, of course, a whole range of applications where extra power may be needed for only a short time. Certain high-speed naval craft and emergency electric generation are obvious examples. Here so-called " boost " turbines can provide a saving in weight, space, and first cost, and the higher fuel bills, because of the short running time, are relatively unimportant.

A more promising application of the gas turbine is perhaps the low pressure end of a compound power plant, the high pressure end being provided by a reciprocating engine supplying hot gases to the turbine, which, because of the relatively low temperature of its working medium, need not be of costly construction. This idea also had its genesis in the aircraft field, the line of development culminating in the Napier Nomad in which both the piston engine and the turbine were coupled to the final drive shaft, and which produced the extraordinarily good fuel consumption of 0·345 lb/equivalent h.p.-hour.[26] A rather special development along these lines was the Orion gas generator turbo compound engine.[27] In this case the reciprocating engine merely supplied hot gas to the turbine, which was the sole source of power. This unit gave a high power/weight ratio for tanks and other tracked vehicles. This arrangement is at the opposite extreme from the turboblown diesel. The power plants are similar in conception, but in the latter case the power is drawn from the reciprocating component, and in the former case it is drawn from the rotating component.

An alternative arrangement of this type of power plant is to use a free-piston engine as a gas generator. The most widely used free-piston gas generator is that emanating from Pescara.[28] The free-piston engine is a two-cycle diesel cylinder in which two pistons, not connected to a crankshaft, are bounced towards one another and driven apart on the firing stroke (Fig 16). The pistons (actually power pistons and compression pistons joined together) shuttle between combustion chamber and air cushion, and in shuttling pump a volume of air for supercharging the engine and to feed a turbine. The Pescara GS34 gas generator of 1250 gas h.p. has been in production in France for some time and is now being made in the U.K.

The overall efficiency of such a compound power plant is 36–37 per cent, some 10 per cent lower than the best slow-speed diesels, but much better than for a gas turbine. The maximum pressures and temperatures can be high in a free-piston power plant because there are no main bearings or connecting rods to consider. This, together with the large excess air and high intake air temperature, is claimed to enable a wide variety of fuels to be burned, but has led to serious piston lubrication problems which are only now being overcome by the use of additive lubricants.[29] The unit has an advantage from the wear point of view in that the position of piston inner dead

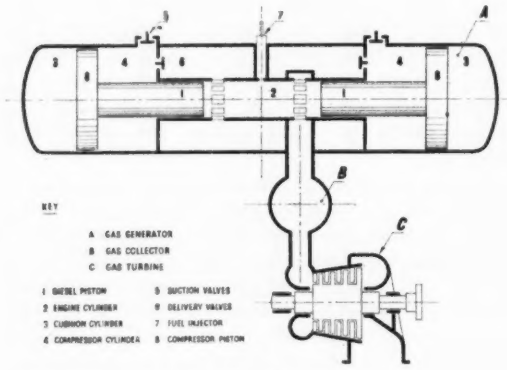

FIG 16

FREE PISTON GAS-GENERATOR TURBINE SET

centre varies with load, and so the formation of a wear step in the cylinder is largely avoided. For this very reason, however, because the shape of the combustion chamber is never twice the same, good fuel distribution is a very real problem, and no less than six injectors are required to give a good combustion in a single cylinder. This would seem to be a potential weakness for running on residual fuel. At full load the inlet temperature to the gas turbine is only 470° C. There is thus no need for costly alloys for the turbine blades, and the unit can run on residual fuels, as the temperature is below that at which sodium sulphate deposits and vanadium corrodes.

The free-piston gas generator with turbine has been used for marine propulsion, for locomotives, electricity generation, and water pumping. It has the advantage that any number of gas generators can be arranged to feed one or more turbines, thus providing flexibility of plant lay-out. With such an arrangement it would be possible to keep the plant running while one gas generator was shut down for repair. Building up a power plant using multiples of one or two standard sizes of gas generators should cheapen each unit and facilitate the supply of spares. A big advantage of the free-piston arrangement is the absence of vibration, enabling a light foundation to be used. A disadvantage is the difficulty of ducting high pressure and high temperature gases.

The fact that the stalled torque of the work turbine is some three times the torque at maximum speed gives flexibility of operation and enables a simpler transmission to be used for traction purposes. This is particularly advantageous for railway traction, where some six locomotives have so far been built, some with free-piston gas generators and some using two-stroke loop scavenge and two-stroke opposed piston engines as gas generators.[31]

Road Vehicle Applications

The application of the gas turbine to road vehicles is being extensively investigated, and there are at least six prototypes being studied in Europe and America having power outputs from 100 to 270 h.p.[30] The main difficulty seems to be the high fuel consumption, particularly at light load and when idling. This could be improved by use of a heat exchanger, either the simple recuperative type as used on the Rover 2S100, or the less bulky rotary regenerative type, which, however, has serious sealing problems which do not yet appear to have been solved. Another way of improving the efficiency at full load would be by raising the permissible inlet gas temperature to the turbine. Here the problem is metallurgical, to develop an alloy which will withstand these severe temperature conditions, but which will not be too expensive to produce. This last requirement involves the ability to cast the compressor rotor and turbine wheel with integral blades. A possible solution which might not require such high duty alloys is the use of blade cooling. Specific fuel consumptions as good as 15 m.p.g. are often quoted for steady speed conditions, but in normal road use the figure would be less favourable.

Most road vehicle gas turbines employ a two-shaft design with a free power turbine, and this free turbine is an effective torque converter, constituting some part of an acceptable automatic transmission. The maximum torque ratio of a turbine is, however, normally no more than 3 : 1, and although this would suffice with one emergency gear for a passenger car, a change speed gear-box is still required in order to obtain the 10 or 12 : 1 necessary for a commercial vehicle. Nevertheless, in any comparison of cost and weight of a gas turbine with a petrol engine, the engine + transmission should always be considered.

There are differences of opinion at the present time as to whether the gas turbine will have its first application to the passenger car field or to the commercial vehicle. Here there is a fundamentally different approach on both sides of the Atlantic. The Americans look first to performance and are impressed by the high power and low bulk and weight of the gas turbine. They see applications for it to heavy lorry and fast coach transport. In the U.K.

and Europe transport companies are very fuel cost conscious, and are already operating with the highly efficient diesel engine. If higher power outputs are required they are more likely to come from the supercharged diesel engine than from the gas turbine. In the European passenger car field the smallest gas turbine so far produced is towards the top end of the power bracket required for present vehicles. There seems no sign of an efficient gas turbine being produced in the size required for the great bulk of the European car market. Here the first application is more likely to be in the high price, high performance car field, where the novelty alone might be sufficient attraction to sell a good number.

The free-piston engine in combination with a gas turbine has been suggested for the passenger car, and prototypes have been produced by two large manufacturers in the U.S.A. The General Motors Hyprex [32] is a siamesed unit containing two cylinders, the diesel portions of which run in a bore of 4 inches diameter. The unit has a maximum operating speed of 2400 cycles/min and is nominally rated at 250 gas h.p. The main advantage for automotive propulsion is claimed to be relative silence and lack of vibration when competing with the diesel engine, and economy and ability to burn cheaper fuel when competing with the gasoline engine.[30] As compared with the simple gas turbine it has the advantage of greater efficiency and a lower gas inlet temperature to the turbine, which would simplify the blade problem, but it would appear to be at a disadvantage so far as weight is concerned. The relatively large bulk of the free-piston engine is due in part to the use of automatic valves which limit the maximum speed of the unit. Whether the combustion difficulties, which require six injectors to overcome on the large unit, will be a problem remains to be seen. General Motors claim to have overcome the high fuel consumption at idling by a system of charge circulation.

Ford's free-piston engine has been applied to a farm tractor.[33] This unit has a power cylinder of 3·75 inches bore and delivers 160 h.p. It is claimed to be much lighter and smaller than an oil engine of comparable power, and the quoted specific fuel consumption is 0·45 lb/g.h.p/hr. The torque characteristics of the turbine are favourable for the tractor application and enable some saving to be made in the transmission.

DISCUSSION

In this very rapid survey of the whole field of petroleum usage it is inevitable that there have been some omissions. Nothing has been said of natural gas or of low pressure gas. Neither of these gases are important sources of energy in Britain at the moment, but they are both very important in the U.S.A. Each year millions of tons of volatile petroleum fuel are run to waste in the Middle East. If experiments on the carriage of this material in refrigerated ships are successful we may expect to see an expansion of its field of usefulness in the U.K.

Shortage of space has also restricted discussion of developments in the lubricating oil field. The new engine and fuel developments have placed a big burden on the lubricating oil, which has to do more now than just lubricate the working parts. The transfer of dispersant, anti-oxidant, and anti-corrosive agents to the affected parts has become an accepted part of their function. In fact, some of the engine developments described would not have been possible without the use of special lubricants. H.D. oils have enabled high-sulphur Middle East distillates to be used in high-speed diesel engines, and very high additive oils, such as those of the emulsion type, have, as mentioned above, quite transformed the marine diesel engine picture. The free-piston engine is evidently providing its own special lubrication problems. In the automotive field the use of low viscosity oils has improved the overall fuel consumption of road vehicles, has given easier starting in cold weather, and a reduction in wear due to a more rapid circulation of oil. The use of ultra high viscosity index multi-grade oils has enabled this to be done without sacrificing lubrication qualities under operating temperature conditions.

Sufficient has probably been said in the earlier sections to show that this is a period of change so far as petroleum fuels and the prime movers which burn them are concerned. The two important fields would appear to be the industrial and the road transport. We are about to see an intensification of competition between the diesel engine, the free-piston compound engine, and the gas turbine for the stationary, locomotive, and marine applications. Where thermal efficiency differences are marginal, the choice of power unit might well be determined by ability to burn cheaper low grade fuel. Thus the free piston gas generator + turbine which at the moment has a thermal efficiency some 10 per cent less than that of the diesel engine could be a strong competitor for the latter unit in most fields if it could demonstrate a superior ability to use low cost fuels. In the locomotive engine field in the U.S.A. the railway companies are relaxing their once rigid specifications. Some railroads are experimenting with blends consisting mainly of residual, with a small percentage of distillate. Others are using a light residual alone. In the U.K. the coming dieselization of the railroads will increase the requirement for distillate fuel, and one could hope that this demand could be channelled to a " B " grade to leave more of the lighter fuel for the high-speed engine.

The increase in demand for middle distillate fuels due in part to the expansion in the use of the diesel engine at the expense of the petrol engine is posing a

serious problem for the oil companies.[34] This is illustrated by the fact that whereas the deliveries of oil fuel to vehicle operators from January to September 1956 were 11·7 per cent higher than in the corresponding period of 1955, the total consumption of petrol rose by only 2·1 per cent and the consumption of petrol by commercial users actually fell.[35] This is not necessarily a bad thing, as from the national point of view it is obviously important that the maximum useful work should be obtained from scarce and expensive imported fuel. This demands that the fuel should be used in the most efficient prime mover for the particular application. The problems posed by this development can best be met, not by mutual recrimination, but by a simultaneous attack by the three parties concerned:

(1) The engine builder—to make his product capable of digesting the widest possible variety of fuel, consistent with maintaining a good thermal efficiency and satisfactory operating characteristics.

(2) The oil company—to be prepared to introduce new refining and marketing techniques to meet the changes in customers' requirement.

(3) The government and local authorities—to become more flexible in their attitude towards the fiscal and storage regulations for the various grades of fuel.

So far as the diesel engine itself is concerned, a considerable amount of work has already been done to widen the range of fuels which it will accept. The development of the larger size of engine to run on residual fuel has already been mentioned. In the smaller sizes military requirements have acted as a stimulus to the development of an all-fuel engine. It has been shown to be quite possible to run most diesel engines fairly satisfactorily on 80 O.N. petrol. Extremely good results can be obtained with some forms of combustion chamber which, however, usually involve some compromise with other aspects of engine performance such as smoke limit and high speed operation. It is a matter of interest to speculate whether the all-fuel engine will remain a military curiosity or whether it will have a commercial application, if not in Europe and America, where the existing grade system is well established, at least in the more remote markets, where it may be uneconomic to attempt to distribute the full range of petroleum products.

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DISCUSSION

A. T. Wilford: My remarks are from the standpoint of the road passenger vehicle operator. The author has drawn our attention to the continuing development of the petrol engine in the direction of higher compression ratios and to the prospect that this engine will soon closely approach the diesel engine in regard to specific fuel consumption. He also suggests that this modern version of the petrol engine will gain some ground from the diesel engine. However, he conceded, and I fully agree, that it will only be in the field of luxury coach travel. Its use in place of the diesel engine in buses, limited stop coaches (such as the Green Line services), and in long distance coaches would seem to be unlikely.

In the first place the operators of these types of vehicles would be reluctant to return to the storage and use of a fuel which is inherently less safe than is derv fuel

so far as the fire risk is concerned. One minor point in this connexion is that it is now quite common for staff to be permitted to smoke in garages where only derv fuel is stored; it is, moreover, a concession which would be very difficult to withdraw.

More important, equality in specific fuel consumption is not the same thing as equality in fuel costs. Specific fuel consumption is related to consumption on a weight basis, whereas fuel is used and paid for—plus tax—on a volume basis. Fuel specific gravity is therefore of prime importance to operators who cannot charge luxury fares and are conscious that each time fares are raised passengers are lost. There is even a demand for a derv fuel of the highest possible specific gravity, qualified only by the requirements that there shall be no adverse effect on cold starting or on exhaust cleanliness. There is a near inverse relationship between m.p.g. and fuel specific gravity. Some two or three years ago London Transport carried out a very large scale test with buses in service comparing a derv fuel of sp. gr. 0·830 with one of sp. gr. 0·848 (these figures being mean values over a period of 10 months). The measured increase in consumption due to the use of the lower sp. gr. fuel was 1·5 per cent, although the statistical precision of the test was such that the increase might have been as much as 2·8 per cent. With a fuel bill of £6 million per year, which is what London Transport spends, even a 1·5 per cent increase in consumption would cost £90,000 a year, with no compensating benefits. The adsorption of a petrol engine with the same specific fuel consumption as a diesel would be likely to increase our fuel consumption by 12 per cent of more. (An increase in fuel consumption of the same order would, of course, arise if petrol MT80, for example, was used in place of derv in diesel engines.)

There is a further objection to the use of such a petrol engine in public service vehicles, this being its inferior performance at part loads and slow speeds. London Transport's red buses have a scheduled speed of about 11 m.p.h., and on some portions of the busiest routes the scheduled speed is only 7 m.p.h. The country buses have a scheduled speed of about 14 m.p.h., while even the Green Line coaches can only be scheduled at about 18 m.p.h. None of the vehicles can exceed the 30-m.p.h. limit, and the great majority of the coach routes pass through the congested central area of London.

The supercharged diesel is of much greater potential interest, and limited tests have already been carried out with both types of blower mentioned by the author. The investigations are still very much in the experimental stage, but Mr Downs gives reasons for suggesting that the positive displacement blower may be found to be the more applicable to our types of service. We are looking to the supercharger as a means for obtaining a reduction in fuel consumption rather than an increase in engine output; we are also hoping that it may further assist our efforts to run with a virtually clean exhaust. Perhaps the author would let us have his views as to whether these objectives are likely to be achieved.

Mr Downs has referred to the use of low viscosity oils, and our pioneer work in this connexion is already well known. The whole of London Transport's fleet is using an engine of SAE 10W viscosity and, compared with the use of an SAE 30 oil which was the standard lubricant a few years ago, we are economizing in fuel at the rate of £350,000 a year. Extensive tests have been carried out with SAE 5W oil, but under the conditions of operation this grade does not give any further improvement in fuel consumption.

London Transport has also been employing additive type oils for some years past, the grade at present in use being of the type covered by the MIL-L-2104 and DEF-2102-A specifications. Large-scale tests have also been carried out with Supplement I oils, but at the time it was not possible to show that these oils led to any increase in the life of engines between overhauls. In the meantime, however, various modifications, combined with improved methods of maintenance, have led to a substantial extension of engine life, and the stage has now been reached once again where the life of the engine is limited by the condition of the piston assembly. We are therefore contemplating further tests with higher additive oils, and have the Series 2 variety in mind.

Our continuous and successful efforts to reduce engine maintenance costs makes us very conservative towards any suggestions for the use of a wider range fuel. If, as might be inferred from the author's remarks on marine diesel experience, the objections to these lower grade fuels can be overcome by the use of lubricating oils of sufficiently high additive content, we should like to see the oils developed and proved before taking any risks. Having regard to the present trend in the road passenger transport industry, where there is a steady loss of revenue arising from competition of the private car, whether it is moving or parked at the side of the road—thereby adding to traffic congestion—no operator can view with equanimity any charge which is likely to increase fuel consumption or engine maintenance costs. The wider range fuels might well give some improvement in fuel consumption, but their effect on engine life is highly suspect.

D. Downs: I agree with Mr Wilford's comparison of the petrol with the diesel engine. Despite the fact that higher compression ratios are now being used in the petrol engine and that its specific fuel consumption at full load is approaching that of the diesel, I do not think that there will be much application of the petrol engine in the commercial vehicle field in the near future. My reference to the luxury coach was really a concession to the petrol enthusiast, indicating the one possible direction in which the petrol engine might make a comeback, albeit in a limited and specialized field.

I was interested in Mr Wilford's remarks about the specific gravity of diesel fuel, because it is the great emphasis that those who are responsible for the purchase of large quantities of fuel for transport companies place on specific gravity which makes it very difficult for the oil companies to dilute the diesel range with lower gravity fuel from the kerosine or petrol fractions. This is a direction in which the road vehicle commercial engine could help the fuel position, not so much by taking in from the heavier fuel end, but by taking in a little from the gasoline range, where there is at the moment some evidence of over-production. The rigid stand which road vehicle operators take on the question of specific gravity is understandable as things stand at the moment, for, as Mr Wilford said, tax is paid on volume. There is perhaps scope for a change in the fiscal regulations which would be helpful to both petroleum supplier and customer.

The supercharger will probably reduce fuel consumption on the bus. It will certainly give a cleaner exhaust and, by using a supercharged engine of small size to do the work of a larger unsupercharged engine, there should be a saving in bulk and weight. In some cases this could mean an increase of one or two in the number of seats which could be put into a bus.

K. Arter: Some American experience would suggest that the high compression car may be capable of rather greater economy than Mr Downs has indicated in his paper, so I thought it might be worthwhile to find out

what the possibilities are. Ten years ago General Motors Research in Detroit built an experimental V8 engine, of 4½ litres displacement. This could be operated at four alternative compression ratios, 6, 8, 10, or 12 : 1. At 8 : 1 the maximum brake horsepower was 136 at 4000 rev/min. Steps were taken to ensure the same volumetric efficiency with each compression ratio and it was found that the mechanical efficiency at full throttle was virtually independent of compression ratio. Performance data on this engine were published in some detail in two SAE papers by Roensch in 1949 and 1950. There appears to be very little published information on more modern production engines and some of my friends in Detroit tell me that the papers published by Roensch are still regarded there as valid.

Taking Roensch's figures, we get the curves shown at the bottom of Fig A. I have expressed the data in terms of pounds of fuel per hour against the brake horsepower of the engine at 3200 rev/min. Curve A is for the 8 : 1 compression cylinder head and Curve B is for the 12 : 1. The horsepower is net, *i.e.* the actual horsepower available at the flywheel with the fan and silencing system connected. The 12 : 1 engine gave a maximum torque 12½ per cent greater than the 8 : 1, so for a strict comparison of fuel economy on the basis of equal car performance (in other words, equal brake horsepower), Curve A should be compared with the 12 : 1 engine running 12½ per cent slower, or 2800 rev/min, *i.e.* with Curve C. If we then compare Curves C and A, there is an almost constant saving of 8½ lb of fuel an hour over the range from 60 h.p. up to the full throttle output of 128 h.p. Of this 8½-lb saving, 6 is due to the higher compression ratio and 2½ to the change in axle ratio.

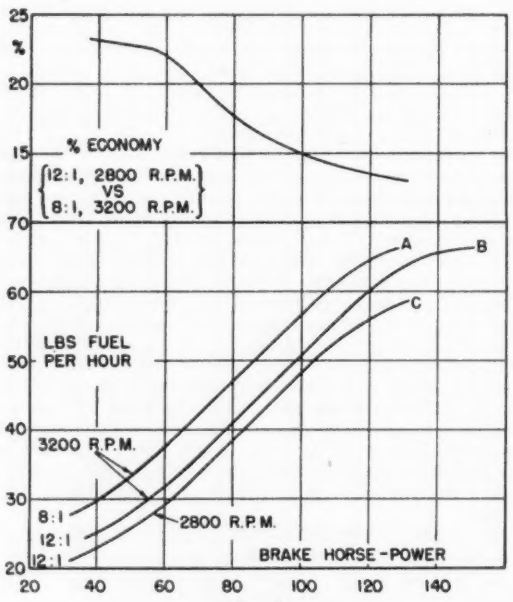

Fig A

Expressing the total saving as a percentage of the consumption at the lower compression ratio, we obtain the curve at the top of Fig A; thus, for equal car performance, there is a fuel saving of 13 per cent at maximum horsepower, rising to 23 per cent at the lightest loads. On the other hand, if we choose to retain the same axle ratio with the higher torque, the fuel saving will be less, and the full throttle acceleration much improved. Because rolling resistance and windage are unchanged, all of the 12½ per cent extra torque is available to overcome inertia and the time to accelerate, say from 15 to 60 m.p.h., might be reduced as much as 20 per cent. Fig B shows a similar

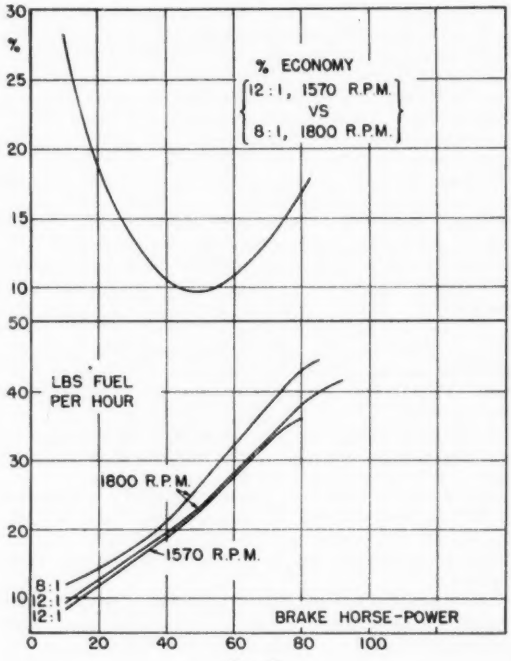

Fig B

comparison at a lower speed, 1800 rev/min. The differences here are not quite so clearly defined. It appears from the shape of the top curve that there was some fault in the carburation of combustion around half load at this speed, but since American cars cruise at a low load factor, maybe 20 to 25 per cent, the car would still be capable of showing an economy of the order of

Table A

Compression ratio	8 : 1	12 : 1
287 in.³ engine: Lb fuel per hour at 55% load factor .	42·3	34·0
96 in.³ engine: Lb fuel per hour	14·1	11·3
Miles per imp. gallon at 55 m.p.h. (sp. gr. = 0·75)	28·8	36·1
Relative m.p.g.	100	125

15 to 20 per cent for the 12 : 1 as against the 8 : 1 compression ratio.

The popular types of European cars operate at higher load factors, but similar economies appear to be quite possible. Suppose a 1½-litre car is taken as an example. This has one-third the displacement of the General Motors engine under consideration and in Table A I have assumed that the fuel consumption can be scaled down in the same proportion. Analysis of some road test data

has indicated that the overall consumption of a 1½-litre car is usually equal to its m.p.g. as measured at a constant road speed of 55 m.p.h. At this speed the engine load factor will be about 55 per cent, and the piston speed about 1900 ft/min, which happens to be the piston speed of the General Motors engine at 3200 rev/min. In other words, we can apply the data from Fig A. Assuming a specific gravity of 0·74, we see at the bottom of Table A that the 1½-litre car will have a consumption of 28·8 m.p.g. at 8 : 1 compression ratio, and at 12 : 1 it goes up to 36·1 m.p.g., a 25 per cent overall improvement in m.p.g.

This sounds very worthwhile and it would justify a higher cost for the high octane fuel than is assumed by Mr Downs. I wonder if he has any data for multi-cylinder engines at alternative compression ratios to show whether my rather theoretical calculations are at all realistic.

Now let us look at the General Motors data from a different point of view. Fig C repeats two of the curves from Fig 1 in Mr Downs' paper, and on the same graph I have drawn the corresponding data for the General Motors V8 engine. The dotted lines show that the brake thermal efficiency of the V8 at 12 : 1 was no better than the figure given by Mr Downs for 6 : 1. I would like to ask him whether he considers his curve for brake thermal efficiency is a realistic one for a well-designed multi-cylinder engine operating with all the usual accessories.

Referring again to American cars, the latest models now in production are running largely at 9½ or 10 : 1

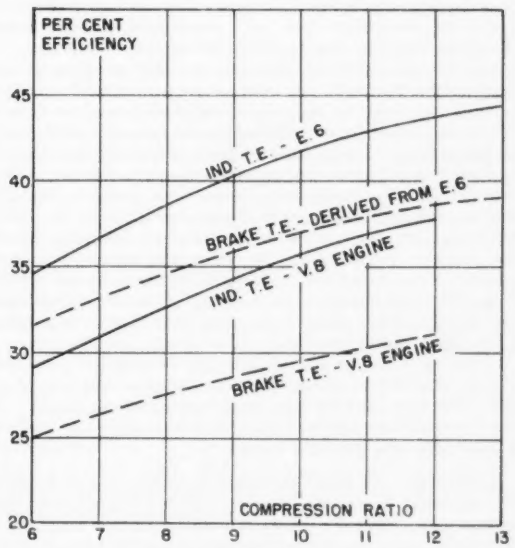

Fig C

compression ratio, but it appears there are quite a lot of public complaints of heavy fuel consumption. That may be partly due to other factors, such as the addition of air conditioning equipment and heavier cars, but I do think that the further rise of compression ratio, although it will go on, may take place at a slower rate over the next year or two.

There are two reasons for this: one is that they have had quite an amount of hot starting trouble, and I gather that some of the starting motors now being fitted develop a greater torque to overcome the hot starting difficulty than would be necessary to start the engine cold at −20° F. The other problem they call "rumble," a trouble which nobody seems to know very much about. It appears that on some of the cars a higher octane requirement is needed to protect the engine against "rumble" than is needed to overcome knock. The trouble is aggravated by a retarded spark, and it does appear that these very high compression ratio engines are getting very critical. The effective range of spark timing is now becoming extremely narrow, so it seems that further increases in the octane numbers of American

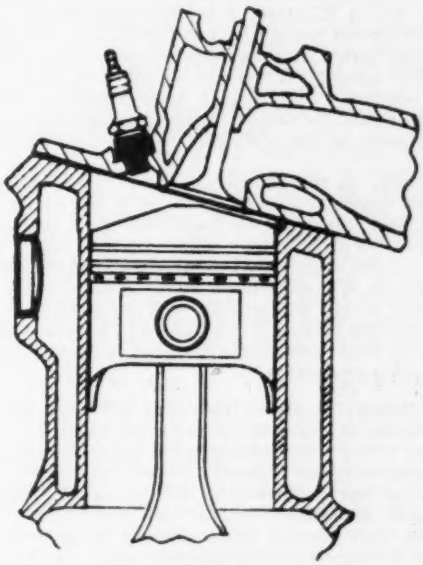

Fig D

fuels may be taken up in catering for these problems, rather than giving greater engine efficiency.

Nevertheless, a 12 : 1 engine may be on the market in the U.S.A. in two or three years' time. It is worth noting that their gasoline trucks are still at the 8 : 1 or 8½ : 1 level, although operators of these vehicles must be concerned with fuel costs and would demand higher ratios if they thought them worthwhile.

On the question of combustion chamber shapes, the latest information just coming over is that there is quite a trend towards putting the combustion chamber entirely in the cylinder block. Fig D illustrates one of the 1958 models, which bears some resemblance to the design in Fig 4 of the paper, except that all the valve mechanism is in the cylinder head. This has a flat machined surface, as shown. The piston has a pent-roof top and the whole of the compression space is contained by machined surfaces within the cylinder block. That enables them to control production tolerances and keep variations of compression between cylinders down to a reasonable limit, something like half a compression ratio.

I fully agree with the author that the hemispherical type of head loses its attraction at the higher ratios for the reason he has given. I am glad he made reference to the need for further improvements in valve material and design. Although the British industry has now adopted XB steel, it continues to lag behind American practice. The shortage of nickel in Germany has forced the Germans to adopt a compromise which would appear to be

capable of withstanding more severe duty than the straight XB valve. I refer to their method of using a silchrome quality valve, but with a ring of special alloy at the face. This alloy is high in nickel and manganese.

I am a little surprised that British cars in Canada and the U.S.A. do not run into more complaints of loss of compression, considering the high lead content in the gasolines there. In view of the small difference in cost, it would be a good insurance on the part of British exporters to follow American practice and use something like 21·4 NS. If they do not, I think they are going to run into trouble sooner or later.

On petrol injection, I believe there is one further reason for the interest in it in the U.S.A. a year or two ago, and that is that the manufacturers are very anxious to get a lower hood line and the injection system offers the possibility of that. I have heard it suggested recently that, for similar reasons, we may find ourselves going back to the horizontal carburettor one of these days.

Finally, on the question of small diesel engines, Mr Downs forecasts increasing use of a common cylinder block and crankshaft for diesel and spark ignition versions of the same basic engine. Does he imply by this that a diesel engine can be built for the same weight per cubic inch of displacement as the gasoline engine? In terms of weight per horsepower, I am under the impression that the diesel will continue to be at some disadvantage, at least until turbo-charging or supercharging is on a mass production basis.

D. Downs: I agree with Mr Arter that Roensch's paper is probably the most up-to-date which has been produced in the U.S.A.; it certainly gives the most information on multi-cylinder engines. I feel that some special pleading has been used in comparing the engines of different compression ratio. Mr Arter says, for instance, that because more power is being obtained at 12 : 1, the engine speed can be reduced and the equivalent power obtained at a lower engine speed. This, of course, gives a more favourable efficiency ratio for the higher compression ratio, but the fact of the matter is that the engine would not be run at a lower speed at the higher compression ratio. It is most likely that the customer would make use of the greater power available or, if the designer was satisfied that the original power was sufficient, the engine capacity would be reduced, in which case I think the efficiency ratios given in my paper would be applicable.

Mr Arter commented on the fact that my brake thermal efficiencies were somewhat higher than those quoted by Roensch. It is some time since I read Roensch's paper, and I am not sure what mixture strength he used. My brake thermal efficiencies were taken at the minimum specific fuel consumptions. His may have been taken at a somewhat richer mixture strength, and I think that is the explanation of the difference.

G. M. Barrett: Mr Downs says that the marine diesel must be capable of burning residual fuel, if not 3500 sec Bunker "C," then the intermediate 1500 sec blend. Although some owners are operating motor ships on marine fuels having viscosities up to 3500 sec R.I. at 100° F, it should be remembered that "C" grade fuel oil is allowed a maximum viscosity of 6500 sec conforming to the now obsolete Bunker "C" specification of the U.S. Navy Department.

Later on Mr Downs is deliberately non-committal concerning the proportion of wear with residual fuel, due to corrosion from the higher sulphur content and to abrasion from ash and hard carbonaceous deposits. Many of us feel that the little laboratory evidence available in favour of abrasive cylinder wear by hard carbon is distinctly dubious and that it is not supported by results in the field. However, the results of many years work at Thornton Research Centre suggest that the overall rate of cylinder wear is the product of the rate of corrosive wear multiplied by the rate of abrasive wear, and is not just the sum of these two quantities. Thus it is possible to cut the corrosive wear down by, say, 70 per cent, by providing sufficiently highly basic additive to combat the sulphur acids and thereby reduce the overall wear rate also by 70 per cent.

Mr Downs rightly draws attention to the awkward shape of the combustion chamber in free piston gas generators, and the wording implies that six injectors are always required to give good combustion in the single cylinder. I am sure that the author would agree that this depends on the cylinder diameter. The GS 34, which has a diameter of 13½ inches, has six injectors, whereas the Muntz CS 75 of 7½-inch bore has only four injectors. The Ford experimental tractor engine of 3¾-inch bore has two injectors, while the General Motors Hyprex of 4-inch bore with Siamesed cylinders has only one per cylinder.

Further on the statement is made that the stalled torque of the turbine is some three times the torque at maximum speed, which lends itself to traction and automotive applications. This figure is rather conservative, as the ratio for the turbine in the first locomotive on the French National Railways is 3·8 : 1.

Mr Downs expresses the hope that, with the increasing dieselization of British Railways, their fuel requirements could be channelled to a "B" grade fuel to leave more distillate fuel for the high-speed engine. However, it must be remembered that under the modernization programme about 20 per cent of the total diesel traction horsepower will be high-speed engines in railcar units. It would create many difficulties if running sheds and motive power depots had to have two separate grades of fuel, one for locos and one for railcars. Furthermore, several of the main-line locomotive designs include engines having crankshaft speeds running up to 1500 rev/min. In view of the necessity to build up from scratch the necessary maintenance and repair organization to service eventually some 3700 diesel locomotives, it would seem wiser not to add to maintenance problems by using "B" grade fuel until experience has been built up on distillate fuel.

Finally, I would like to ask Mr Downs his personal views on a matter which was very much to the fore soon after the war, the surface discharge of spark plugs. A lot of people have great hopes of this for burning weaker mixtures in the gasoline engine.

D. Downs: As far as the wear in marine diesel engines is concerned, I was deliberately non-committal as to the relative importance of abrasive and corrosive wear, as I feel that much more experimental work will have to be done before the final answer is obtained. Undoubtedly both are important, and I would agree with Mr Barrett that they should be multiplied rather than added, and that we must do our best to reduce both of them as much as possible.

The torque range of 3 which I quoted for the gas turbine is, I agree, conservative, and it can be increased by taking special measures. I was quoting a figure for the normal range of gas turbines, but I am interested to learn that the French obtain 3·8 on their locomotive gas turbine.

I agree that British Railways would probably be well advised to commence operation of diesel engines using

distillate fuel. Once they have their repair and maintenance facilities established, however, economic circumstances will force them to consider using as cheap a grade of fuel as the engine will digest. Although this may involve the use of two grades of fuel, one for the main engines and one for the auxiliary engines and small shunting locos, I think they will find that the provision of the extra storage and dispensing facilities will be justified by the fuel cost saving.

It is some years since I have done any work on a surface discharge spark plug. The last time I experimented with one was on a small two-stroke engine, which normally could not run for more than 30 hours before the spark plug failed by whiskering or deposit fouling. With one type of surface discharge plug we had a trouble-free run of over 500 hours duration. The insulator was, however, severely eroded in this period. This is the fundamental difficulty with the surface discharge plug. The semi-conducting material which is normally used as the insulator is not strong enough mechanically to withstand engine operating conditions. To increase its strength, it is customary to blend in other materials. It is difficult to do this, however, and still retain the semi-conducting properties. This is the dilemma which, so far as I know, the spark plug manufacturers have not yet surmounted.

E. H. Wild: Could I ask Mr Downs whether the compression ratios he quoted for the U.S.A. and the U.K. were weighted averages or arithmetic averages? The type of average would make quite a difference in drawing various conclusions between the rates of increase in the different countries.

The other query I would like to put is the trend in the U.S. compression ratios and, again, the possible reasons why similar trends may not be followed in the U.K. Could it be possible that the basic reason for the rapid increase in the compression ratio of U.S. cars is due to the fact that they have gone the route of the V8 engine, which is an inherently stronger design, and therefore capable of increases in compression ratio without too many further alterations? In other words, when the basic design of the V8 engine was laid down some years ago, they more or less laid the foundations for going up to compression ratios of 10, 11, and 12 : 1, which we are now talking about. In the U.K. the standard engine is either a four- or six-cylinder in line, and to get the maximum benefits from higher compression ratios we may become involved in rather more extensive design changes than are necessary in the U.S.A.

One more reason why we may not go to the higher compression ratios to the same extent as the U.S.A. is the dependence of our motor industry on the export markets. We shall therefore be dependent for some time on the availability of higher quality fuels in our major export markets.

D. Downs: The British figures for average compression ratio were arithmetic averages of car types. The American figures, which were taken from the annual survey published by *Automotive Industries*, have, I believe, a similar basis.

I think that the point about the strength of the American V8 engine is a good one. A further factor is one which I mentioned in the paper, which is that the American engine, being large for the duty which it has to do, probably operates at a lower load factor than the British engine, and so it is in fact operating for only a small proportion of the time under conditions which could give high maximum pressures. I do not think that the majority of British in-line engines would need much strengthening to enable them to use a compression ratio of 10 : 1. As Mr Wild says, the export market may be a reason why our manufacturers are conservative about compression ratio.

C. D. Brewer: I would like to refer to the last contribution on this question of compression ratio in America. I can give assurance that the $10\frac{1}{2}$: 1 is genuinely used, at least by the company with which I am familiar. There are several important reasons for using this high ratio in the U.S.A. First, it can be done because of the availability of high octane fuel and of the predominance of automatic transmissions. Secondly, and perhaps the main reason, is that now there is general agreement between motor companies that they should not advertise maximum performance, there is a tendency to quote maximum torque, and they have been able to show that by increasing compression ratios to $10\frac{1}{2}$: 1, b.m.e.p. at quite reasonable speeds, say 3000 rev/min, of about 166/175 SAE rating or 1·15 lb.ft/cu. in. of capacity are obtained. I would endorse also what Mr Wild said about compression ratios used in engines of vehicles produced in the U.K. These vehicles are primarily for export, and since it is still necessary to cater for quite large territories where regular grade fuels of perhaps 75 to 78 O.N. only are available, this does rather restrict the manufacturing company's initiative in regard to the use of higher compression ratios, although there may also be other reasons for doing this.

I would also like to endorse what Mr Downs said regarding engine weights for increases in compression ratio. I do not think that, provided the basic design is adequate, there is any practical limit to compression ratio within the scope of present normally marketed fuels, and there is no need to increase the present weight of engines.

Would Mr Downs be good enough to amplify the figures for b.m.e.p. for increasing compression ratios he showed for the petrol and diesel engines? I thought that the shapes of the curves, particularly that for the petrol engine, appeared a little unusual. I would say that, from my own experience, the curve for the diesel engine is unusual in that the increase in m.e.p. above 10 : 1 to about 15 : 1 was shown to be increasing, apparently linearly. The values for the petrol engine seemed very pessimistic, and those for the diesel decidedly optimistic.

D. Downs: It is true that this curve can be argued about. Its virtue is that it is a comparison between the petrol and the diesel engine obtained on the same crankcase, but using different cylinder heads. The compression ratio with the diesel head was fixed, but with the petrol head was varied from 5 to 13. Such a comparison is, in one sense, a little unfair on the petrol engine because the volumetric efficiencies of both versions were identical over the speed range, whereas in the practical case one could certainly obtain a higher volumetric efficiency with the petrol engine because of the greater freedom which one has in the disposition of the valves and in the use of valve overlap. On the other hand, the diesel engine shows up well on a power basis because a form of diesel combustion chamber was used which enabled over 95 per cent of the available air to be consumed. With a more conventional direct injection engine, the power output would not be so good. It is better, therefore, not to place too much emphasis on particular figures in this comparison, but to look at the broad picture presented by the curves.

Dr W. T. Lyn: On the question of petrol *v.* diesel, the mode of ignition will have no direct influence on the

thermodynamics of the cycles. In fact, due to the generally higher rate of heat release in a diesel engine, there is more constant volume burning in the diesel than in the petrol engine. For the same compression ratio, therefore, one cannot expect the petrol engine to be more efficient, even if fuel of high enough octane number is available.

On the matter of wear, the relative contribution between abrasive and corrosive action must depend on the type of fuel and engine used. My own experience with residual blends of fuel on small high-speed engines has shown that abrasive wear is the dominating factor and that lubricating oil additives, while helping to maintain a cleaner engine, do little in combating the wear.

With regard to the unified crankcase and cylinder block for petrol and diesel heads, several engine manuturers have admitted in a recent informal discussion that the result of such a practice invariably led to the withdrawal of the petrol version, although the latter was introduced first. The general view was in entire agreement with Mr Downs' survey. It is clear that, except in the field of the passenger car, the diesel engine almost monopolizes transport and industrial applications. Even in the passenger car field the major difficulties, noise at light load and idling and the additional cost involved, will be gradually overcome. The position as regards the mechanism of diesel engine noise and the technical aspect of noise reduction are becoming clear, and the problem is becoming one of compromise with cost. My own belief is that the kind of outrageous clanging from diesel vehicles which one sometimes hears on the road is entirely unnecessary and that the popularity of the diesel car will come sooner than is generally expected.

From the fuel requirement point of view, again diesel engines appear to be the most natural form of prime mover, since they can digest the largest percentage of oil from straight distillation of the crude without any special treatment. In fact, it is so omnivorous that we even expect it to digest special fuels which were developed for the petrol engine and produced in such large quantities as to be left over. It looks as if the diesel is the only existing engine that fulfils the requirements given in the concluding section of Mr Downs' paper, and I should like to know if he agrees. In this connexion, a " premium " grade diesel fuel has been mentioned earlier in the text. I should like to ask the author's view of the requirement of such a grade of fuel. High cetane number, I agree, can help little as regards light load noise, but I wonder whether it will help to alleviate the problem of misfiring at light load and high speed.

D. Downs: For equal compression ratios, the diesel engine will still show to advantage over the petrol engine because of its better efficiency at part load. In the diesel engine the thermal efficiency increases as the load is reduced because of the lower average temperature of the cycle. In the petrol engine the hot residuals have an unfavourable effect on efficiency at light load.

I am afraid that I disagree with Dr Lyn about abrasion being the major factor in wear, even with a residual fuel. If wear is due to abrasion, how can the phenomenal effect of emulsion lubricants be explained?

Although some progress has been made in the study of noise in the diesel engine, I would not go so far as Dr Lyn in suggesting that technically it has been solved. An improvement in noise level usually involves a compromise with smoke limit and maximum power. We cannot yet have our cake and eat it. We can have a quiet engine, or a smokeless engine, or a high-powered engine, or a low fuel consumption engine, but we usually have to decide which is most important for a particular application, concentrate on that factor, and be content to have the others in smaller measure.

I do not think that a case has yet been made out for a premium fuel. High-speed misfiring can be a problem, and it can be tackled either from the fuel angle or from the fuel injection equipment angle. At the moment, I am inclined to think that a form of variable timing device on the fuel injection pump would be the biggest single factor in alleviating trouble.

We have experimented with a fuel the cetane number of which was increased by ten numbers above the normal 52–55 by the use of an ignition accelerator. The higher ignition quality of this fuel enabled the injection timing to be retarded without experiencing misfiring at high speed, but even with this advantage it was found that the improvement in noise level at idling and at light load was only just detectable. There would not, therefore, seem to be much case for a premium fuel having an O.N. higher than 55.

S. T. Walker: The author says that the use of low viscosity oils has led to a reduction in wear through more rapid circulation of the oil. Presumably he is considering here the wear that takes place on the rings and liner in the first few minutes of running, which has been found, in certain circumstances, to be the major wear taking place on these components.

At the Research Station of the British Petroleum Company work on the examination of engine wear has been carried out for some years with radioactive tracers. By this means it is possible to determine the wear which takes place in those first vital minutes of operation.

Experiments have been conducted to assess the effect of oil viscosity on wear at start-up. The temperature of the oil in the engine sump was adjusted to give a 10 to 1 variation in viscosity at the time of starting the engine, but despite this wide difference only a slight change in wear was detected.

It appears from this work that it is not simply viscosity which controls wear, but that other aspects of an oil's formulation play a major part.

D. Downs: I agree with Mr Walker's remarks on lubricating oil formulation. Space limitations did not permit me to develop the subject of lubrication as I had at first intended and so my remarks under that heading are necessarily somewhat brief and condensed.

L. A. Cooper: I have seen somewhere that specific fuel consumption is minimum for a jacket temperature of about 180° F. I wonder if Mr Downs can comment whether this is valid or not, and whether recognition of this has been taken in the consumption figures presented.

D. Downs: In the experimental work on the E.6 engine described in the paper no attempt was made to obtain the absolute minimum specific fuel consumption or maximum efficiency, because all the results were comparative, being carried out using the same crankshaft and crankcase assembly. In general, the specific fuel consumption decreases as the temperature of the oil is increased because the viscosity of the lubricant, and hence the friction loss, is being reduced. If the viscosity is reduced too much, however, wear and even friction are liable to increase, due to the presence of boundary lubrication conditions rather than full fluid lubrication.

Dr G. B. Maxwell: We have heard a great deal from the engine designers as to what they want, and from the consumers as to what they want to pay for it, but the refiners are the people who have to produce the fuels at an economic rate.

There are two points in this connexion. One is this question of the higher compression ratios in the U.S.A. as compared with Britain. The general market demand conditions in the U.S.A. are quite different from those in the U.K. Something like 40 per cent of the total petroleum product sold in the U.S.A. is in the motor spirit range, whereas in the U.K. it is probably of the order of about 25 per cent. At the present time, the only way of producing a high yield of motor spirit from the barrel of crude—in other words, getting more out of it than there is already in the crude—is by catalytic cracking, which has the great advantage of turning out at the same time a very high octane blend stock. Thus, in order to meet the market in the U.S.A. that the oil companies have to supply, they are almost automatically at the same time turning out a high octane fuel. This is rather different from Britain, where it is desirable, if possible, to cut down the motor gasoline we are turning out from a barrel of crude.

For this reason, we should probably welcome a reduction in the amount of catalytic cracking, which involves a conversion of part of the fuel oil—of which there tends to be a shortage—into high O.N. motor spirit to meet the requirements of the modern high-compression spark-ignition engine.

We have had outlined to us two ways in which this "octane number race" might be halted. One is to extend the use of the compression-ignition engine, possibly slightly modified so as to run without loss of efficiency on a wide distillation range fuel. The second is to retain both compression and spark ignition engines but to modify the latter, possibly by development of petrol injection, so as to give good efficiency on lower O.N. fuels.

D. Downs: I do not think that there is much chance of petrol injection so improving the light load running of the spark ignition engine that the part load thermal efficiency will approach that obtained on the diesel engine. It has been done on a sleeve valve engine where the orderly movement of the air permits a rich mixture to be placed in the vicinity of the spark plug to give stratified operation at light load. Despite a great deal of experimental work, it has not been found possible to do this on a poppet valve engine. The fundamental difficulty is that there is so much indiscriminate movement of air within the combustion chamber that the pattern of fuel distribution imposed at the time of injection is inevitably disturbed by the time the spark passes. There have been various attempts to surmount this difficulty by the use of ante-chambers around the spark plug to contain the rich mixture, but so far none of them has been entirely successful.

A case can be made out for using a wider cut diesel fuel. Even this solution is not as simple as it appears at first sight, as Mr Wilford has told us. I suppose that while some oil companies are able to meet the strict requirements for gravity imposed by the transport undertakings, this will act as a brake on the broadening of the range of fuel type which can be described as diesel fuel. If the diesel fuel supply position became desperate and all the oil companies were equally affected, the transport undertakings would have to accept some widening of the diesel cut, even though this involved some reduction in gravity.

G. J. Parsons: Two previous speakers have referred to the use of Series II (or III) oils; Mr Wilford with some hope, and Dr Lyn with some disappointment. This prompts me to emphasize that if maximum advantage is to be obtained from the use of HD oils, they should be selected to suit the operating conditions. A paper to the Institution of Mechanical Engineers earlier in 1957, by Dyson, Richards, and Williams, dealt with this point by showing how an oil can be selected from the range of HD oils available so as to ensure that performance is satisfactory, particularly with regard to cleanliness in the ring belt area.

In Mr Wilford's particular case it is difficult to generalize, because it is well known that his operating conditions are a little unusual; for example, he is one of the very few operators whose engines stick to lower rings in preference to top rings. That remark is not meant to imply that the higher performance oils, such as Series III oils, will not be of value in his case; I would recommend, however, that to avoid disappointment Mr Wilford keeps clearly in mind the purpose of such oils when embarking on his trial.

Several speakers have referred to the matter of abrasive and corrosive wear. I suggest that a third mechanism of wear, namely attritious wear or metal-to-metal contact, should not be forgotten. It is becoming evident that with engine outputs continually climbing, particularly in the diesel field following turbo-charging, ring lubrication conditions are becoming very severe, and this type of wear may occur. This problem is, perhaps, a good example of how both engine manufacturer and oil supplier must co-operate in finding a solution.

R. J. Murchie: Has Mr Downs found any advantage with multigrade oils when compared with some of the 10W oils?

D. Downs: I cannot give any comparative figures from road tests comparing a light oil with a multigrade oil. On general grounds one might expect the multigrade oil to perform better. My only doubt on this point arises from the fact that the multigrade characteristic is obtained by adding a V.I. improver to a basically thin oil. We know that an oil containing a V.I. improver is a non-Newtonian fluid, and that consequently its viscosity will decrease with increase in rate of shear. Its viscosity, therefore, in those parts of the engine where the lubricant is subjected to a high rate of shear, is largely an unknown factor. This is my principal doubt concerning high V.I. multigrade oils and I have not seen sufficient information published on this subject to allay my doubts.

The meeting then closed with a unanimous vote of thanks to the author.

THE INFLUENCE OF ALKALI ON THE OXIDATION OF LUBRICATING OILS *

By K. U. INGOLD † and I. E. PUDDINGTON †

SUMMARY

The oxidation of hydrocarbon lubricating oils produces acids, ketones, esters, small amounts of alcohols and lactones, and traces of anhydrides, together with an increase in the number of conjugated double bonds. An oil-insoluble sludge is also produced, probably by a series of esterification reactions involving difunctional oxidation products, chiefly hydroxy acids, followed by a " drying " action similar to that observed in the drying oils. In the presence of alkali the rate of oxidation of lubricating oils is increased, the oils are partially bleached, and sludge and varnish formation is totally suppressed. This last effect is due to the neutralization of the organic acids, which prevents the formation of the polyester sludge and varnish precursors.

INTRODUCTION

One of the major problems involved in the use or storage of mineral oils in any system open to the atmosphere is the eventual deterioration of the oils owing to oxidation. The products of oxidation are acidic and tend to corrode their container and, in addition, products are formed which undergo transformation by polymerization and condensation into a semi-solid " sludge " insoluble in the oil. To prevent the formation of acids and sludge, oxidation inhibitors, which are generally phenols, amines, or sulphur compounds are frequently added to oils. Corrosion inhibitors, which form a protective coating on metal surfaces, may also be added; these inhibitors are generally organo-metallic compounds of phosphorus, arsenic, or antimony. Various alkaline compounds have also been proposed as acid neutralizers and as sludge and corrosion inhibitors.

Alkaline reagents are frequently used in petroleum refining to neutralize the acidic constituents normally present in the oil and those added to it during refining.[1] It has been common practice to remove the alkaline reagents by washing with water so as to leave the oil neutral and with a low ash content, but there would seem to be no doubt that some of the improvement in oil stability achieved by this process is due to residual traces of alkali left in the oil. For instance, it has been realized for some time that weak bases, such as magnesium oxide, have an inhibiting effect on the oxidation of transformer oils,[2] and that alkali and alkaline earth soaps have an anti-sludging action on lubricating oils.[3] However, it is only comparatively recently that the advantages in maintaining a high alkalinity in the oil have been fully realized [4–7] and that attention has been devoted to the addition of basic compounds to oils. Some of the elements and compounds that have been proposed as additives include metallic sodium and its alloys; [8–10] metallic calcium; [9–11] metallic magnesium; [10,12] Group II metals, hydroxide, and basic salts; [11,13–15] alumina; [16] alkali, alkaline earth and ammonium phosphates, arsenites, arsenates, and bismuthates; [17] highly basic alkaline earth alkylarylsulphonates; [18] and sodium alkyl phenates.[19] It has also been proposed to bleach oils by oxidizing in the presence of strong bases such as metallic sodium [20] or in the presence of alkali metal hydroxides and activated carbon.[21]

At first sight, it might appear that bases act solely as corrosion inhibitors, *i.e.* by neutralizing organic acids they prevent the dissolution of heavy metal oxidation catalysts such as iron or copper. However, it has been found that bases can inhibit sludging and lacquering even in the absence of metallic surfaces,[22] and the present work, which describes a study of the influence of strong bases on the kinetics and products of lubricating oil oxidation, was started with the hope of explaining these effects.

EXPERIMENTAL

Most of the work was done with a semi-finished oil SAE 30 (viscosity at 100° F = 588 S.S.U., viscosity index 88, average formula $C_{29}H_{55}$, iodine number 13·4) and with a premium grade commercial SAE 10 oil that had been used in an automobile without an oil filter, under normal driving conditions, for 2500 miles.

Kinetic measurements of oxygen uptake were made on an apparatus similar to that of Cooper and Melville,[23] except that provision was made for a continuous circulation of the oxygen over the oil and through a cold trap in order to remove volatile products such as water. It was found that unless this precaution was taken the volatile products concentrated above the oil and eventually prevented further oxidation. The reaction vessel was shaken rapidly to ensure that the measured rate of oxidation was not determined by the rate of dissolution of oxygen in the oil.

Potentiometric titration curves were measured on a Fisher Recordomatic Titrimeter; they were made with an N/20 *iso*propanol KOH solution on oil samples

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dissolved in a 1 : 1 mixture of benzene and *iso*propanol. The starting pH and acid number (with the end point taken as pH 11·0) of the oxidized oils were obtained from these curves. The transparencies of the oils are reported as the percentage of red light ($\lambda = 6200$ Å) transmitted by a Lumitron Colorimeter (Photovolt Corpn, New York), the instrument being adjusted to give a transmission of 100 per cent with white medicinal oil. Infra-red spectra were taken with a Model 21 Perkin Elmer infra-red spectrophotometer using a sodium chloride prism.

RESULTS

Kinetic Measurements

Measurements were made on the rate of O_2 uptake by the neutral semi-finished SAE 30 hydrocarbon oil containing no additives. The kinetics were not studied very thoroughly, as the chief object was to obtain samples of oil which had reacted with known volumes of oxygen. Oils oxidized in the presence and in the absence of strong bases are called hereafter " alkaline oxidized oils " and " normal oxidized oils " respectively.

In Fig 1 the rate of O_2 uptake by the SAE 30 oil at 190° C is plotted against the time for both the oil alone and the oil plus 1 per cent sodium hydroxide. This quantity of alkali was sufficient to cause a very pronounced catalysis, increased amounts having no additional effect. Alkalis also increased the rate of oxidation of a number of other lubricating oils and reduced the induction periods of highly refined medicinal oils. Rates of O_2 uptake under different conditions were compared by measuring the time taken for a given volume of O_2 to be absorbed. Arrhenius plots of the reciprocal of the time taken for the SAE 30 oil to react with 5 cc O_2 (at S.T.P.)/cc oil are shown in Fig 2.

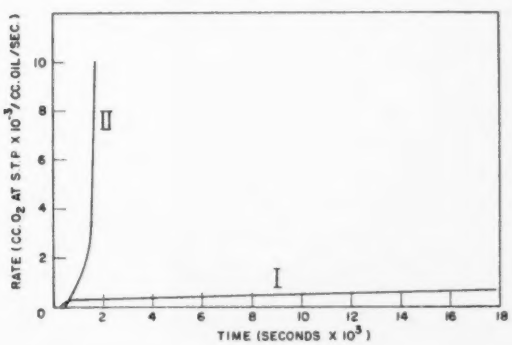

FIG 1

OXIDATION OF THE SEMI-FINISHED SAE 30 OIL AT 190° C
I. Normal oxidation II. Alkaline oxidation (1% NaOH)

Solid metal catalysts were found to have a pronounced effect on the rate in the absence of alkali but very little effect in its presence, *e.g.* 1 per cent of metallic copper with a surface area of 0·08 cm²/cc oil increased the rate of the alkaline oxidation (1 per cent NaOH) by 30 per cent and of the normal oxidation by 200 per cent.

The most remarkable effect of a strong base on the oxidation of lubricating oils is its bleaching action.[20,21] Table I gives the percentage transmission of red light

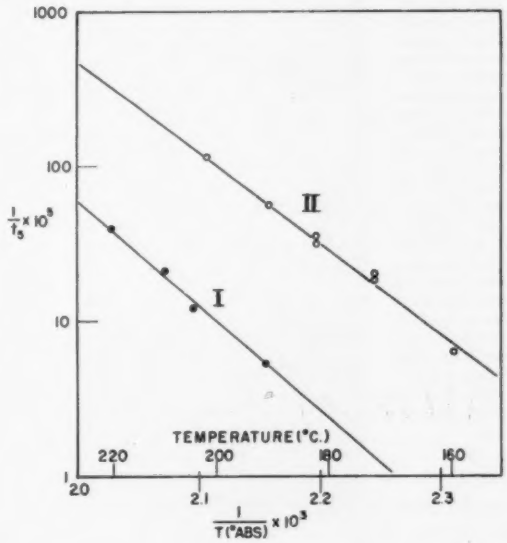

FIG 2

ARRHENIUS PLOT OF THE TIME TAKEN BY THE SAE 30 OIL TO REACT WITH 5 CC O_2 AT S.T.P/CC OIL (t_5)
I. Normal oxidation II. Alkaline oxidation (1% NaOH)

for the semi-finished SAE 30 oil oxidized to varying extents both in the presence and absence of NaOH. The table also includes some measurements on a sample of the same oil fully finished to meet API service requirements M.S. and D.G. and on the alkaline oxidized oil treated with water and with dilute hydrochloric acid after oxidation. The transparency of the oil in a normal oxidation was reduced to zero by the addition of as little as 1·0–1·5 cc O_2/cc oil, but in the presence of 1 per cent NaOH (added as a single pellet) as much as 37 cc O_2/cc oil actually increased the transparency. A minimum in the transparency was observed during oxidation in the presence of NaOH, which did not occur when metallic sodium was used.[20] For the same degree of oxidation, the temperature of the reaction had little effect on the oil colour. Washing alkaline oxidized oils with water also generally had little effect on their colour, but washing with dilute mineral acids darkened them. The liberated organic acids or their reaction products must, therefore, be darker in colour than the salts from which they were derived.

The bleaching effect was not confined to unoxidized oils. In one experiment the oil was oxidized with

TABLE I

Properties of an Oxidized SAE 30 Oil

Oil	Additive	cc O_2 at S.T.P/cc oil	Treatment	% transmission	pH	Acid No.	Viscosity S.S.U. at 100° F
Semi-finished	None	0·0	None	34	9·3	0·19	588
Semi-finished	None	0·5	None	3·1	4·9	0·40	604
Semi-finished	None	1·0	None	0·9	4·6	0·50	612
Semi-finished	None	2·0	None	0	4·4	0·62	626
Semi-finished	None	5·0	None	0	4·1	0·93	672
Semi-finished	None	14·5	None	0	3·9	2·23	785
Semi-finished	1% NaOH	0·5	None	47	8·5	0·36	603
Semi-finished	1% NaOH	1·0	None	48	7·8	0·40	613
Semi-finished	1% NaOH	2·0	None	12	6·4	0·57	616
Semi-finished	1% NaOH	2·5	None	16	6·5	0·71	627
Semi-finished	1% NaOH	5·0	None	68	7·9	0·92	696
Semi-finished	1% NaOH	14·5	None	71	7·9	2·90	1620
Semi-finished	1% NaOH	37·0	None	59	8·0	7·32	2960
Semi-finished	1% Na	5·0	None	72	—	—	—
Semi-finished	10% NaOH	5·0	None	61	10·7	0·17	823
Semi-finished	1% Cu	14·5	None	0	3·9	2·47	733
Semi-finished	1% Cu + 1% NaOH	14·5	None	10	8·1	2·00	844
Semi-finished	1% NaOH	1·0	Water wash	46	7·9	0·29	633
Semi-finished	1% NaOH	2·0	Water wash	24	6·3	0·46	643
Semi-finished	1% NaOH	5·0	Water wash	29	4·7	1·03	707
Semi-finished	1% NaOH	14·5	Water wash	55	6·2	3·19	845
Semi-finished	1% NaOH	37·0	Water wash	60	6·8	5·67	1150
Semi-finished	1% NaOH	14·5	Acid wash	5	3·7	2·96	810
Fully finished	None	0·0	None	85	—	—	—
Fully finished	None	0·5	None	45	—	—	—
Fully finished	None	1·0	None	0	—	—	—

2 cc O_2/cc oil in the absence of alkali (0 per cent transmission), 1 per cent NaOH was then added, and the oil was oxidized with a further 5 cc O_2, which produced sufficient bleaching to give a 20 per cent transmission.

Measurements of pH and acid number on alkaline oxidized oils were made on the oil decanted from the NaOH pellet, but small amounts of abraded NaOH may still have been present. The pH's of the oxidized oils showed the same general features as the transmissivities, a low pH corresponding to a dark oil. In the absence of alkali there was a large decrease in pH for the first 0·5 cc O_2, followed by a much slower decrease for further oxidation. In the presence of NaOH the pH showed the same minimum around 2 cc O_2 as was observed with the transparencies. The acid numbers were found to give only a moderately good measure of the degree of oxidation, whether or not alkali was present. Some of the curves from which the acid numbers and pH's were derived are shown in Fig 3. Oxidation of the oil increased its viscosity, particularly in the presence of NaOH with more than 5 cc O_2/cc oil, owing to the formation of insoluble, but well dispersed, soaps.

Effect of Alkali on Sludge and Varnish Formation

Sludge and varnish formation generally occurred together, so that an oil oxidized with as little as 0·5 cc O_2/cc oil contained a dark brown or black sludge as a colloidal suspension, together with a brown varnish or lacquer attached to the walls of the vessel. No dark coloured sludge or varnish was, however, formed under basic conditions, although for high degrees of oxidation orange-coloured soaps were sometimes precipitated from the cold oil.

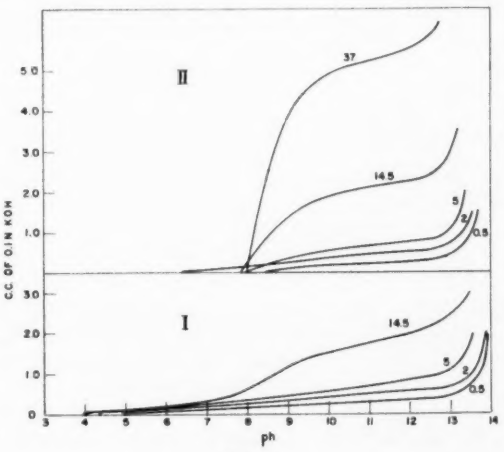

FIG 3

POTENTIOMETRIC TITRATION OF 4 G SAMPLES OF OXIDIZED SAE 30 OIL

I. Normal oxidation II. Alkaline oxidation (1% NaOH)

The numbers on the individual curves denote the number of cc O_2 at S.T.P. adsorbed/cc oil.

Some qualitative experiments were made with the SAE 10 oil that had been used in an automobile for 2500 miles. Samples of this oil were filtered and

heated with various additives at 150° C for 28 hours. The oil without additives gave a large amount of sludge, but the sludging was not due solely to further oxidation, since a sample in a sealed evacuated tube also sludged slightly. Prior contact bleaching with 12 per cent activated clay inhibited sludging, and so did the addition of 10 per cent of a commercial heavy duty concentrate and 10 per cent of the soaps separated from an oil oxidized in the presence of NaOH. All strong bases tested prevented sludge formation provided that they were intimately mixed with the oil. The result of slowly stirring single pellets of sodium metal in 50 cc of the oil at 115° C for 24 hours is given in Table II. It can be seen that less than 0·1 per cent of sodium was sufficient to suppress sludge and varnish formation and to maintain the pH, even though the unheated oil was quite acid. It should be noted that although the sodium

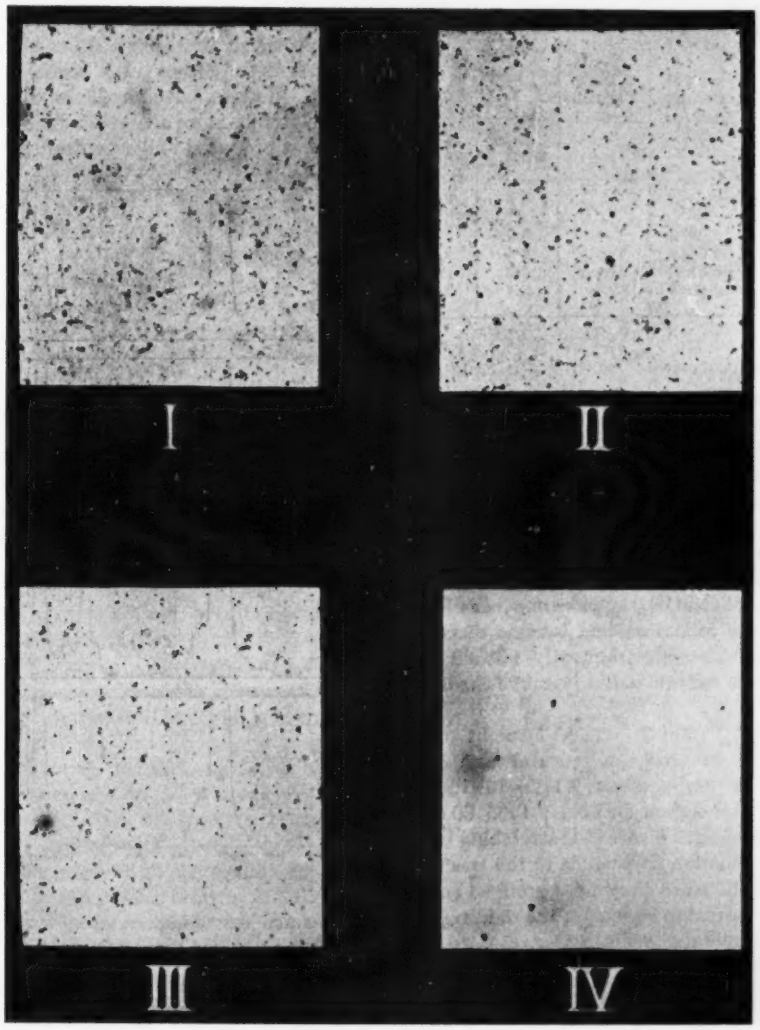

FIG 4

PHOTOMICROGRAPHS (× 200) OF 50-CC SAMPLES OF THE FILTERED USED MOTOR SAE 10 OIL AFTER OXIDATION FOR 24 HOURS AT 115° C WITH ADDED SODIUM

I. No sodium
II. 5 mg Na
III. 10 mg Na
IV. 40 mg Na

was not emulsified in the oil, the slow stirring continuously exposed a clear surface. Photomicrographs of the oil after oxidation are shown in Fig 4.

TABLE II

Sludge Formation in 50-cc Samples of the Used SAE 10 Oil Oxidized for 24 Hours at 115° C

Weight of Na before oxidation (g)	Weight of Na after oxidation (g)	Weight of sludge produced (g)	Varnish formed	Potentiometric titration of filtered oil	
				Starting pH	Acid number
Original oil				5·3	2·45
—	—	0·0680	Yes	3·7	1·96
0·0050	0·0000	0·0550	Yes	4·5	2·16
0·0100	0·0008	0·0380	Trace	5·6	2·11
0·0400	0·0068	0·0050	No	8·7	2·03

Infra-red Measurements

A large number of measurements were made of the infra-red spectra of oxidized oils. The assignments which were made for the various new bands which appear in an oxidized oil are listed below.

O–H stretching (hydrogen bonded, broad); 3650–3100 cm^{-1}, alcohols; 3000–2400 cm^{-1}, carboxylic acids. It should be noted that care had to be taken in interpreting the 3600–3400-cm^{-1} region of an oil oxidized in the presence of NaOH, since some alkali was abraded during the oxidation and became dispersed in the oil. These oils were frequently washed with water or with dilute mineral acid to remove the bands due to NaOH.

C=O stretching; 1830–20 cm^{-1}, straight chain anhydrides; 1780–60 cm^{-1}, anhydrides and/or γ-lactones; 1750–35 cm^{-1}, esters; 1720–1695 cm^{-1}, aliphatic ketones and carboxylic acids; 1565–60 cm^{-1}, ionized carboxylic acids. These C=O stretching bands were the most distinctive new bands in the spectrum of oxidized oils, but since they were grouped sufficiently closely as to overlap seriously, the oxidized oils were treated with alkalis, acids, water, etc., to help identify the products.

C=C stretching; 1605–1595 cm^{-1}, aromatic and conjugated double bond compounds.

The bands below 1500 cm^{-1} were difficult to assign with any certainty, since the oxidized oils generally showed broad absorption from about 1450–800 cm^{-1}. However, the following tentative assignments were made by studying changes in the spectra after washing the oxidized oil with various reagents. Carboxylic acids 1415–05, 1290–1230, and 940–900 cm^{-1}; ionized carboxylic acids 1420–10 cm^{-1}; alcohol 1160–1000 cm^{-1}; ester 1200–1160 cm^{-1}. Inorganic carbonate could also sometimes be detected at 880–75 cm^{-1}.

Oxidation of the semi-finished SAE 30 oil formed acids, ketones, esters, small amounts of alcohols and lactones, traces of anhydrides, and increased conjugation. The spectrum of the unoxidized oil and of the oil oxidized with 14·5 cc O_2/cc oil is given in Fig 5.

In the presence of 1 per cent solid NaOH (added as a single pellet in order to keep its surface area small) and with gentle shaking of the reaction vessel, only the stronger acids produced by oxidation were ionized. The spectra showed that ketones, alcohols, esters, and lactones were present, in addition to the ionized and un-ionized acids, but anhydrides were absent. There was relatively more alcohol and less ester and lactone than was produced under neutral conditions, the most

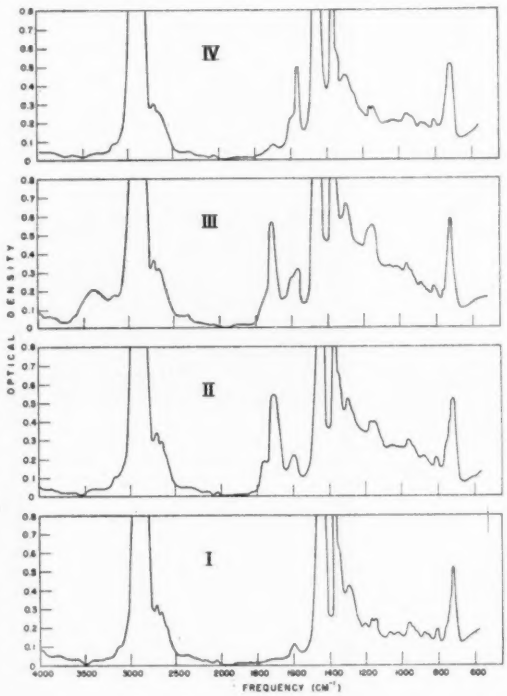

FIG 5

INFRA-RED SPECTRA OF OXIDIZED SAE 30 OIL IN 0·1-MM CELL, REFERENCE CELL EMPTY

I. Unoxidized oil
III. Normal oxidation (14·5 cc O_2/cc oil)
III. Alkaline oxidation, mild conditions, 1% NaOH (14·5 cc O_2/oil)
IV. Alkaline oxidation, very basic conditions, 2% Na metal (8 cc O_2/cc oil)

noticeable difference in the spectra being the increase in the 3400 cm^{-1} band (see Fig 5). By using larger amounts of base, or more vigorously mixing the oil and base, or by using stronger bases, such as sodium metal, the acids were completely ionized and no esters, lactones, or anhydrides were formed. The remaining 1705 cm^{-1} band was therefore due to ketone, and its height indicated that only about 0·1 ketone group was produced per carboxylic acid group, which appeared to be somewhat less than was produced under milder

basic conditions. There also appeared to be rather less alcohol produced, although more was formed than in a normal oxidation. A spectrum for such strongly basic conditions, but with the inorganic bands removed, is given in Fig 5. For both weak and strong bases the concentration of conjugated double bonds in the oil decreased slightly on oxidation.

The ionized acids could be completely removed from an alkaline oxidized oil by washing with water. The 1700-cm^{-1} band was either not affected or else showed a small increase which was attributed to a partial hydrolysis of the soaps. Washing an alkaline oxidized oil with dilute mineral acid left the majority of the organic acids in the oil phase, but the exact changes observed in the spectrum, apart from the disappearance of the 1565–60 cm^{-1} band, depended on whether the oil was extracted with hot or cold acid. Treatment with cold acid gave mainly un-ionized carboxylic acids with little ester, lactone, or anhydride and the oil darkened only slightly. However, if this treated oil was warmed or if the alkaline oil was extracted with warm acid, it got much darker in colour and the anhydride, lactone, and, to a lesser extent, ester bands increased while the acid and alcohol bands decreased. Treatment of this hot acid extracted oil with NaOH completely decomposed the anhydrides but only partially decomposed the esters and lactones.

Acids were prepared from the soaps of alkaline oxidized oils by washing the oils with hot water, acidifying the aqueous phase, and extracting with ether. The acids, called hereafter " Acids A," were dark-coloured liquids which readily condensed to a more viscous product even at room temperature. Their spectrum (Fig 6) shows that they contained mainly acid and alcohol functional groups, together with small traces of ester which may have been formed during the evaporation of the ether. By heating the acids under vacuum at 100° C for 2 hours, the alcohol and acid bands were reduced, with the formation of large amounts of anhydrides and rather less ester and lactone. The greater amount of lactone produced by these acids when dissolved in oil is merely due to the diluent action of the oil promoting intra- rather than inter-molecular reaction. Some analytical data on Acids A are given in Table III, and their titration curve is shown in Fig 7. Potentiometric titrations on the substances mentioned in Table III were made with an N/20 *iso*propanol KOH solution on samples dissolved in a 10 : 10 : 1 mixture of benzene, *iso*propanol, and water. Starting pH's, acid numbers (at the neutralization pH, if any), and acid equivalent weights were obtained from these curves. Saponification numbers and saponification equivalent weights were obtained by refluxing the sample for 3 hours in a 1 : 1 mixture of benzene and N alcoholic KOH, followed by titration with N/2 HCl. The hydroxyl numbers were obtained by the method of Ogg, Porter, and Willitts.[24]

A dark brown or black oil insoluble sludge (iodine number 21) was prepared by oxidizing the semi-finished SAE 30 oil. It was separated by filtration, washed with petroleum ether, and extracted with chloroform. The first traces of sludge formed in the oil were completely chloroform-soluble, but as the

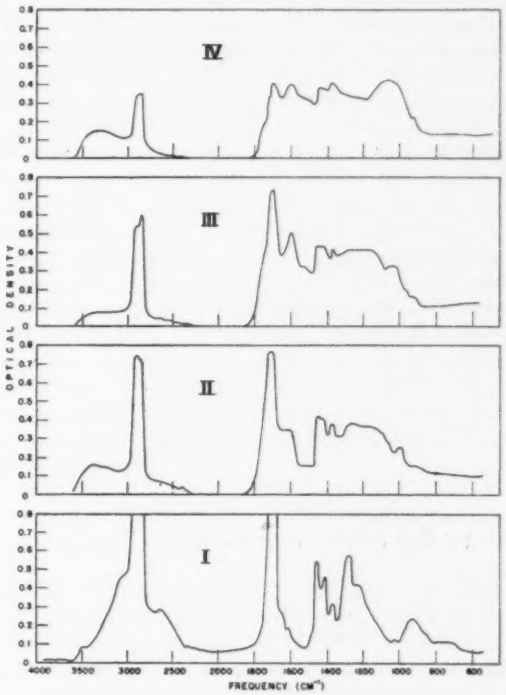

FIG 6

INFRA-RED SPECTRA OF OIL OXIDATION PRODUCTS IN 1·0-MM CELL, RUN AGAINST SOLVENT IN REFERENCE CELL

I. " Acids A " from SAE 30 oil in chloroform (0·155 g/cc)
II. " Acids I " from SAE 30 oil in chloroform (0·13 g/cc)
III. " Residue " from SAE 30 oil in chloroform (0·13 g/cc)
IV. Benzene soluble sludge formed in car from SAE 10 oil (0·09 g/cc)

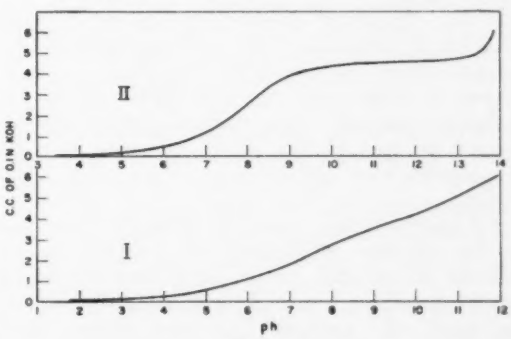

FIG 7

POTENTIOMETRIC TITRATION CURVES FOR SAE 30 OIL OXIDATION PRODUCTS

I. 1·4 g chloroform soluble sludge. II. 0·10 g Acid A

oxidation proceeded an increasing fraction of the sludge could not be dissolved even by boiling chloroform. The infra-red spectra of a number of chloroform-soluble sludge samples showed the presence of hydrogen-bonded acids and alcohols. Esters, lactones, and, probably, ketones were also present, but there was very little anhydride (Fig 6). The spectra also showed a broad absorption from 1500 to 800 cm^{-1}. By shaking a sample of sludge dissolved in chloroform with solid NaOH at room temperature, the quantity of ionized acid and alcohol was increased at the expense of acids, esters, and γ-lactones.

It has been shown recently (R. D. Offenhauer, J. A. Brennan, and R. C. Miller, *Industr. Engng Chem.*, 1957, **49**, 1265) that aromatic thiols are largely responsible, after oxidation to sulphonic acids, for sediment formation in catalytically cracked distillate Acids II were hard gummy substances which gave a deep red solution in chloroform at the same concentration. Acids III and the residue were hard brittle resins which formed black opaque solutions. The acids and residue were dissolved in chloroform and their infra-red spectra measured; that of Acids I is shown in Fig 6. Acids II, Acids III, and the residue had rather similar spectra, the main difference being a slight reduction of the O–H and C–H stretching bands from Acids II to Acids III, to the residue, and an increase in the band due to conjugated double bonds. For convenience, because of the minor differences in these three spectra, only the spectrum of the residue is given in Fig 6. Its spectrum is very similar to that of the unfractionated sludge. Additional data on the fractionated and unfractionated sludge are given in Table III and Fig 7.

TABLE III

Analysis of Oil Oxidation Products

Oil	Sludge	Formula	Starting pH	Acid No.	Titration equiv. wt	Sapon. No.	Sapon. equiv. wt	Hydroxy No.
SAE 30	Unfractionated, $CHCl_3$-sol	$C_NH_{1\cdot33N}O_{0\cdot15N}$	1·8	28	2000	110	509	0·16
SAE 30	Acids I	$C_NH_{1\cdot44N}O_{0\cdot21N}$	3·3	72	780	120	468	0·60
SAE 30	Acids II	$C_NH_{1\cdot44N}O_{0\cdot13N}$	4·0	70	802	130	432	0·25
SAE 30	Acids III	$C_NH_{1\cdot27N}O_{0\cdot15N}$	3·2	54	1038	—	—	0·16
SAE 30	Residue	$C_NH_{1\cdot20N}O_{0\cdot15N}$	3·5	No turning point in titration curve		—	—	0·15
SAE 30	$CHCl_3$-insols	$C_NH_{1\cdot15N}O_{0\cdot23N}$	—	—	—	—	—	—
SAE 30	Acids A	$C_NH_{1\cdot70N}O_{0\cdot20N}$	3·4	218	257	250	224	0·07
Fuel Oil	$CHCl_3$-sol	$C_NH_{0\cdot99N}O_{0\cdot19N}$	3·1	104	539	—	—	—
Used Auto. Oil:								
SAE 10	$CHCl_3$-sol	$C_NH_{1\cdot32N}O_{0\cdot24N}$	5·0	No turning point in titration curve		—	—	4·5
SAE 10	C_6H_6-sol	$C_NH_{1\cdot26N}O_{0\cdot29N}$	4·8	107	527	174	322	2·5
SAE 10	Vacuum	$C_NH_{1\cdot38N}O_{0\cdot26N}$	4·3	133	422	—	—	—

fuel oils. The sludge obtained from the semi-finished SAE 30 oil contained 1·8 per cent sulphur, but the infra-red spectra of the soluble portions of the sludge showed no evidence for sulphonic acids. In this oil, therefore, although some sulphonic acids may be formed and may catalyse sludge formation, they do not constitute an important part of the acids in the sludge.

The purified sludge was separated into four fractions by the procedure of Denison and Clayton,[25] which consists of extracting with hot alcohol to give a filtrate containing "Acids I," and insolubles that are saponified with alcoholic KOH and filtered. This filtrate, after acidification, contains "Acids II"; the insolubles are then acidified and extracted with alcohol to give "Acids III" and a "Residue." The three acids generally constituted 10–25 per cent of the chloroform-soluble sludge. Acids I were soft gummy products which readily esterified and, at a concentration of 0·13 g/cc, gave a red solution in chloroform.

Measurements were also made on the sludge accumulated in the SAE 10 oil during 2500 miles of normal service. The oil contained about 2 per cent solids which were separated by filtration and washed with petroleum ether. When this sludge was extracted with chloroform or benzene in a Soxhlet apparatus, it disintegrated and diffused through the thimble, although it was shown by centrifuging that only about 8 per cent of it was actually soluble in chloroform and that even less was soluble in benzene. The insoluble portion consisted mainly of lead, present largely as lead chloride, with lesser amounts of lead bromide and lead oxybromide ($PbO{\cdot}PbBr_2$). The total lead, calculated as $PbCl_2$, amounted to 70 per cent of the sludge; the balance was organic matter with a C : H ratio of 1 : 1·25, together with iron (1·2 per cent) and spectroscopic quantities of copper and barium.

The spectrum of the motor oil filtrate showed the presence of acids, alcohols, esters, and, probably,

ketones; it indicated a degree of oxidation of about 1·0 cc O_2/cc oil. The small extent of oxidation of the filtered oil is in agreement with the results obtained by Ventsel,[26] who found that the spectra and molecular weights of filtered used oils from a number of I.C. engines were nearly identical with the unused oils. The spectrum of the chloroform- or benzene-soluble sludge which was formed in the car engine (Fig 6) shows the presence of alcohols, acids, esters, and lactones. The spectrum is generally similar to that of the sludge produced by oxidizing the SAE 30 oil in the laboratory at a much higher temperature, except that the car sludge contained considerably more hydroxyl groups. The sludge produced by heating the filtered oil under vacuum was similar to the sludge formed in the car.

The sludge which formed in a comparatively unsaturated (Iodine Number 25) domestic fuel oil after storage for a few months at room temperature had, as would be expected, a lower C : H ratio than the lubricating oil sludges. It contained acids, esters, and alcohols, and its spectrum was similar to that of the car sludge. On the other hand, the spectrum of the sludge prepared by oxidizing the fuel oil at 140° C was more similar to the laboratory-prepared lubricating oil sludges. The sludge precursors were not as soluble in light fuel oil as in lubricating oils.

DISCUSSION

Although alkalis were oxidation catalysts for all of the oils studied in this work, it is known that they can also act as inhibitors with certain hydrocarbons, *e.g.* paraffin wax.[27] The actual mechanism of catalysis or inhibition is not known, but presumably in one case alkali promotes the formation of free radicals, and in other cases either destroys the free radicals or prevents their formation. It should, of course, always be remembered that oxidation *per se* is not necessarily bad for an oil; whether it is or not will depend on whether the oxidation products are harmful.

Acid numbers have in the past often been used to measure the degree of oxidation of an oil. However, a plot of acid number against the volume of oxygen absorbed was concave towards the oxygen axis, particularly below 2 cc O_2/cc oil. This meant that the relative proportion of acidic oxidation products decreased with increasing oxidation and that care should be used when estimating the degree of oxidation indirectly.

From a practical point of view, the pH of an oil is of much greater importance than its acid number.[7] Table I shows that an alkaline oxidized oil maintains a high pH even at large acid numbers, *i.e.* the oil remains non-corrosive. A normal oxidized oil, on the other hand, reaches a low pH and becomes corrosive while its acid number is still small (see Table I and Fig 3). Sludge has higher acid and saponification numbers than the oil in which it is formed, and therefore the acid and saponification numbers of a used oil are reduced by filtration.[28] It has been observed that the formation of sludge may, in fact, be sometimes accompanied by an actual decrease in acidity[28] (see also Table II).

The infra-red spectra showed that acids, alcohols, esters, ketones, γ-lactones, and anhydrides are products of oil oxidation and components of sludge. Separation of the sludge into several fractions and analysis of these fractions showed that they contained acids with more than two oxygen atoms per molecule, a part of the excess oxygen being present as hydroxyl groups. It seems probable, therefore, that the major sludge precursors are "oxy-acids" containing hydroxyl groups. The formation of such difunctional oxidation products, even at low degrees of oxidation, suggests that they must be produced from the initially formed peroxy radical. A probable mechanism would be an intra-molecular hydrogen abstraction to give a hydroperoxide radical,

$$\mathrm{R{-}\underset{\displaystyle O{-}O\cdot}{\underset{|}{CH}}{-}CH_2{-}CH_2{-}R' \longrightarrow R{-}\underset{\displaystyle O{-}O{-}H}{\underset{|}{CH}}{-}CH_2\dot{C}H{-}R'}$$

followed by further oxidation to give a difunctional product. Intra-molecular hydrogen abstractions of this type will particularly favour 1 : 3 oxidation where the geometry of the reaction is most favourable. The formation of γ-lactones is thus readily explained.

Hydroxy acids can undergo various dehydration reactions, whilst inter-molecular reaction will give esters and anhydrides and intra-molecular reaction will give lactones and probably unsaturated compounds. The inter-molecular esterification will, in general, involve more than two molecules of acid and will lead to the formation of chloroform-soluble, hydrocarbon-insoluble, linear polyesters. As the oxidation progresses, the molecular weight of the polyesters will increase both by linear growth and by cross-linking with other polyester molecules to give chloroform-insoluble polymers. Since the oxy acids composing the sludge are highly unsaturated, cross-linking probably occurs mainly by a "drying" mechanism similar to that observed in the drying oils. This is supported by the analytical data in Table III, which show that the insoluble sludge contains relatively more oxygen than the soluble.

The spectra of the normal oxidized oils showed that they contained increased numbers of conjugated double bonds, which was in agreement with the visual observation that the oils darkened rapidly on oxidation and is consistent with the low C : H ratios in the sludge. The mechanism by which conjugated systems are formed probably depends on the fact that the oxidation of unsaturated compounds predominantly occurs α to the double bond[29]; subsequent intra-

molecular dehydration could then lead to the development of a conjugated system. At all events, the darkening of lubricating oils and the formation of dark-coloured sludges would appear to be associated with the unsaturated components of the oils, since fully saturated hydrocarbons do not generally darken appreciably on oxidation. For example, paraffin wax can be oxidized until it contains 50 per cent or more of fatty acids with very little discoloration or polymer formation.[30] Moreover, it was found in the present work that a highly refined medicinal oil would absorb 10 cc O_2/cc oil with only a 4 per cent loss of transparency and without sludge formation, although its spectrum showed that both esters and lactones were produced. This seems to provide further evidence that sludge is formed by prolonged oxidation and "drying" of unsaturated polyesters produced from the hydroxy acids. The formation of cross-linked polymers would also explain why the sludge is not completely destroyed by treatment with alkali, as would be expected if the sludge were entirely polyester.

The inhibition of sludge and varnish formation by strong bases can now be readily explained. Neutralization of the carboxylic acids produced by hydrocarbon oxidation prevents the formation of the esters which are the major components of sludge and varnish. However, many of the data obtained in this work suggest that the action of bases is not confined to a mere neutralization reaction. For instance, the bleaching action of bases must be the result of oxygen attack directed at or near conjugated double bonds with probable C–C bond scission. Moreover, Acids A are more saturated than the sludge acids, and it can therefore be concluded that bases reduce the tendency which the oxygenated products otherwise exhibit of undergoing further oxidation to more unsaturated compounds.

Various interesting speculations can be made from the data in Table III about the oxy-acids that are the major components of sludge. For instance, the formulæ of the saponification products of Acids I and II are $C_{27\cdot6}H_{39\cdot7}O_{5\cdot8}$ and $C_{28\cdot0}H_{40\cdot4}O_{3\cdot6}$ respectively, which suggests that the major sludge precursors are oxy-acids with 1 or 2 fewer carbon atoms/molecule than the original oil ($C_{29}H_{55}$), *i.e.* the acids are produced by oxygen attack promoting bond scission between the α and β carbon atoms, and to a lesser extent between the β and γ atoms, of the hydrocarbon chain.[31] A rather less important sludge precursor also seems to be formed by random oxygen attack on the hydrocarbon molecule to give acids containing an average of 14 or 14·5 carbon atoms/molecule. Acids I, II, and III, the residue, and the chloroform-insoluble sludge are presumably formed progressively by a succession of esterification reactions involving both the C_{28} and the smaller units. Unfortunately sufficient data are not available to confirm this suggestion, and the complete elucidation of the successive steps in sludge formation will undoubtedly be delayed until many pure hydrocarbons have been studied.

The low equivalent weight of Acids A is due to the fact that the low molecular weight acids cannot evaporate from the oil during oxidation in the presence of alkali. Acids I would, therefore, be expected to contain 14·5 carbon atoms/molecule, irrespective of the position of oxygen attack. The measured formula, $C_{13\cdot3}H_{22\cdot5}O_{2\cdot7}$ is in good agreement with theory.

The yields of oil-insoluble sludge and chloroform-soluble products obtained from the used motor oil are in agreement with the results of earlier work on I.C. engines run under moderately low temperature conditions.[32,33] The infra-red curves and C–H analyses obtained in the present work support the suggestion [32,33] that the chloroform-soluble fraction of the sludge was formed by oil oxidation and the balance (90 per cent) was due to the blow-by products of fuel oxidation (*i.e.* partially-oxidized fuel, particles of soot, and inorganic salts derived from fuel additives) together with minor amounts of engine wear products and dust. The spectrum of the chloroform-soluble sludge showed that it contained relatively more oxygen in hydroxyl groups and less in ester and carboxyl groups than the benzene-soluble sludge. This was confirmed by the hydroxyl numbers and by potentiometric titration. The car sludge has a higher pH than the sludge prepared in the laboratory, which is in agreement with the earlier observations that low temperature sludge is neutral,[33] whereas high temperature sludge or varnish is very acidic.[25,33] Raising the temperature of oil oxidation therefore increases the relative amount of C–C bond scission, as would be expected. The low equivalent weight of the vacuum sludge formed by heating the filtered motor oil in the absence of oxygen suggests that it may have been formed by condensation of low molecular weight, oil-soluble, blow-by products.

It has been suggested that piston ring lacquer in I.C. engines arises from a condensation reaction on hot surfaces either of the polymerizable components of partially oxidized fuel (mainly C_8 hydroxy acids) [33] or of the hydroxy acids produced by oil oxidation.[25] The distinction between the two sources of hydroxy acids is probably quite artificial, as they will behave in the same way and, in the majority of engines, they will be formed from both sources. It has been shown [25,34] that oxy acids begin to deposit when their concentration rises above their miscibility in oil (0·2 per cent). This agrees with the present observation that sludge is deposited after the absorption of about 0·5 cc O_2/cc oil (which corresponds 0·3 per cent acids of formula $C_{28}H_{51}O_3$ formed by the reaction

$$C_{29}H_{55} + 3O_2 \longrightarrow C_{28}H_{51}O_3 + H_2O + HCO_2H)$$

and that the filtered used motor oil showed a similar degree of oxidation. The hydroxy acids from the blow-by in an I.C. engine can also be dissolved by the oil. The deposition of sludge and varnish from this source can therefore be reduced by the neutralizing action of bases contained in the oil. This is, in fact, the mechanism of operation of many anti-lacquering oil additives.[25]

Finally it is interesting to note that there is a close similarity in the C : H : O ratios for a number of sludges, varnishes, gums, etc., formed by the oxidative deterioration of a variety of petroleum products.[1,25,32,33,35] It can therefore be concluded that there is an underlying unity to all oxygen-induced low temperature hydrocarbon polymerizations, but a complete understanding of the mechanisms involved will have to await a detailed study of many pure hydrocarbons.

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EVALUATION OF ANTI-SEIZING AND RECOVERY-FROM-SEIZURE PROPERTIES OF E.P. LUBRICANTS BY THE FOUR-BALL TESTING MACHINE *

By C. PALEARI,† A. GIRELLI,† and C. SINIRAMED †

SUMMARY

The paper sets forth and discusses the results of tests made with the four-ball E.P. testing machine by a method which enables one to assess the anti-seizing as well as the recovery-from-seizure properties of E.P. oils. The method tests the oil only in the range of applied loads at which seizure occurs and where the E.P. additive becomes effective.

The results cover a series of samples obtained by adding to the same base oil E.P. additives of different composition (compounds of sulphur, chlorine, phosphorus, and lead soaps). The best results were achieved with sulphur additives, which do, nevertheless, reveal a certain extent of corrosivity on metal surfaces.

THE mean Hertz load (M.H.L.) value alone does not suffice for a stringent evaluation of lubricant E.P. characteristics.

In effect, a low seizing load but high welding load oil (*i.e.* one with good recovery capacity after seizure) could be classified more or less equal to an oil with a high seizing load but low recovery capacity. The latter lubricant may in practice be preferred, for certain uses, to the former oil. In plant working under high but unvarying loads, high seizing load oils will obviously be preferred, whereas in plants subject to considerable sudden overloads (*e.g.* marine propeller reduction gears) a high recovery capacity after seizing may rate first in selecting the oil.

These considerations have persuaded the authors that for any E.P. evaluation of an oil it is necessary to know the minimum seizing load (hence the maximum specific load borne without seizing), as well as the recovery capacity after seizing at high loads.

The determination of the first seizing load should provide a guide to the oiliness characteristics of the oil. The first seizing load does, in effect, tally with the minimum load required to break the film existing between the contacting surfaces and is, to a great extent, connected with the forces with which the lubricant molecules adhere to the metal surfaces.

The recovery-from-seizure properties could instead be represented by an " E.P. index number " obtained from the average of several $\frac{LDH}{x}$ ‡ values determined at loads above the minimum seizure load.

Runs carried out at loads below the first seizure load, where lubricating conditions are still far from the critical point, are of no practical value, in the authors' opinion, in the evaluation of an E.P. lubricant.

The " E.P. index number " computed from the average value of the " corrected load " corresponding to ten runs performed (including that of the first seizure load) should give the true assessment of the tested oil's load-carrying capacity. This figure represents the load required to produce a 1 mm diameter scar on the lower balls, and is expressed in kg. The higher this value is, the greater is the protection against seizure given by the lubricant. This value cannot be obtained for oils which do not contain E.P. additives. In such a case, indeed, it is practically impossible to continue runs above the first seizure load.

The breakdown of the oil film without recovery-from-seizure engenders formation of a very large scar, and despite an appreciable decrease in the pressure because of increased area of contact, which permits lubrication to be established again, friction remains extremely high.

From the above considerations the authors have developed an accelerated method which differs from the M.H.L. determination carried out according to the different procedures proposed [1-4] in that, as seen, it disregards runs made at loads below the first seizing load.

TEST PROCEDURE

(1) *Ascertainment of Minimum Seizing Load*

A 10-sec run is made under a 100-kg load, followed by runs at increasing loads (112, 126, 141, etc.) until the first seizure occurs. If seizure occurs under a load lower than 100 kg, other 10-sec runs are made at decreasing loads (89, 79, 71, etc.). It has been thought wise to start runs at a somewhat low load (100 kg) so that even if non-E.P. oils are handled, the machine will not be unduly stressed owing to immediate seizing without recovery.

* MS received 24 October 1957.

† Stazione Sperimentale per i Combustibili presso il Politecnico, Milano, Italy.

‡ The " corrected load " is expressed by $\frac{LDh}{x}$ where: L = applied load, kg; Dh = Hertz line diameter, mm; x = measured diameter, mm.

Having identified the minimum seizing load, this value will be confirmed if, repeating the run a second time, seizing does actually occur as expected.

(2) *Determination of "E.P. Index Number"*

Having ascertained the minimum seizing load, a further nine runs are made under increasing loads (using the same sequence of loads as for the M.H.L. test).

The series of runs can be performed without reaching welding load, or may be stopped by this load being reached before completion of the series. In the latter case other data are obtained until all nine runs have been made, using intermediate loads (see Table I), ranging from the first seizing load and the welding load.

TABLE I

Applied load kg	*LDh*	Applied load, kg	*LDh*
178	87·40	261	145·6
185	92·03	271	153·1
193	97·37	*282*	161·6
200	102·2	293	169·9
208	107·6	304	178·4
216	113·1	*316*	188·0
224	118·7	329	198·3
233	125·2	342	208·8
242	131·6	*355*	219·4
251	138·3		

This is the case with oils characterized by a high seizing load (about 200 kg), where actually the interval between initial seizing load and welding load is generally small. The mean of the $\frac{LDh}{x}$ values for the ten runs performed represents the "E.P. index number." This value appears more influenced by changes in the oil's E.P. qualities than does the M.H.L. value. Table II shows the values ascertained for seven E.P. oils of different composition tested with the two methods.

TABLE II

Oils[3]	B	C	D	E	F	G	H
M.H.L.	79·4	68·5	71·6	69·8	41·4	71·2	77·3
E.P. index number	89·1	70·5	82·5	75·0	36·2	52·8	79·6
Min. seizing load, kg	158	100	141	126	71	193	158

It can be seen that oil G reveals very clearly its poor recovery capacity, together with an excellent ability to withstand seizing.

To check the reliability of this method of evaluation, the E.P. qualities of some oils, containing additives from among those commonly used, were tested.

This was also intended to give a series of practical values which could be used to rate the E.P. qualities of an oil within a determined range of values.

The additives examined were based on organic compounds of chlorine, phosphorus, sulphur, and metal soaps. Particular attention was turned to the action of a metal soap alone and combined with sulphur; that of chlorinated paraffins; the combined action of sulphur and chlorine; and of sulphur with chlorine and phosphorus.

The additives were tested dissolved in a quantity of 8 per cent by weight in a base oil, consisting of a blend of solvent refined oil and bright stock. The base oil characteristics are reported in Table III.

TABLE III

Specific gravity at 15°/4° C	0·894
Flash point (P.M.)	256° C
Engler viscosity at 50° C	20·7
Engler viscosity at 100° C	3·6
Viscosity index	102
Conradson carbon	0·44%
Ash	Traces
Organic acidity (neutralization number)	0·02 mg KOH/g

The following oil samples were tested:

(1) base oil without additive;

(2) base oil + lead naphthenate (32% Pb);

(3) base oil + lead naphthenate + 0·5% elemental sulphur;

(4) base oil + chlorinated paraffin (40% Cl);

(5) base oil + additive A, containing sulphur (8·5%), chlorine (26%), phosphorus (0·55%);

(6) base oil + additive B, containing sulphur (5·15%), chlorine (21%), phosphorus (0·33%);

(7) base oil + additive C, containing sulphur (58%) and chlorine (29·4%).

Of these, sample (3) was prepared by adding the required quantity of sulphur flowers at about 120° C.

The test results are given in Table IV, and Fig 1 gives the wear–load diagram. As can be observed in Table IV, on some samples more than the established ten tests were made. This was done to observe the performance of the oil under higher loads also. The "E.P. index number" has nevertheless been computed from the average of the first ten tests in ascendant order.

From the results obtained the following considerations can be deduced:

(1) With oil containing no additive the assay cannot be continued beyond the first seizing load. This fact confirms that the evaluation of E.P. additives should be carried out at load levels above the minimum seizing load. It is actually impossible to assess the anti-seizing properties of any oil without operating under seizure conditions.

(2) Just as there is, for every oil and every lubricating system, a temperature at which the oil breaks down,[5] so there is a temperature at

TABLE IV

Load, kg	Oil 1		Oil 2		Oil 3		Oil 4		Oil 5		Oil 6		Oil 7	
	x, mm	$\frac{LDh}{x}$	x, mm	$\frac{LDh}{x}$	x, mm	$\frac{LDh}{x}$	x, mm	$\frac{LDh}{x}$	x, mm	$\frac{LDh}{x}$	x, mm	$\frac{LDh}{x}$	x, mm	$\frac{LDh}{x}$
89	0·478	72·6												
100	2·020	20·0			0·496	81·6	0·459	88·2						
112			1·212	38·9	0·530	89·0	0·756	62·4						
126			1·424	38·8	0·563	98·1	0·852	64·8					0·477	115·7
141			1·452	44·0	0·639	100·1	0·876	73·0	0·776	82·5			0·477	134·2
158			1·570	47·5	0·639	116·8	0·908	82·2	0·832	89·6			0·573	130·5
178			1·730	50·5	0·682	128·1	1·036	84·3	0·965	90·55	1·110	78·9	0·603	133·9
(185)			1·781	51·6					1·032	89·1				
(193)			2·038	47·8					1·932	50·4	1·610	605		
200			1·831	55·9	0·898	113·9	1·905	53·7	2·020	50·5	1·700	60·1	0·776	131·9
(208)			1·870	57·5					2·026	53·1				
(216)									2·143	52·8	1·705	66·4		
224			2·630	45·1	0·861	137·8	2·145	55·3	2·115	56·1	1·695	70·0	1·187	100·0
(233)									2·208	56·7				
(242)									2·250	58·5	1·822	72·1		
251					0·847	163·2	2·275	60·9	2·365	58·5	1·832	75·6	1·234	103·6
(261)														
(271)														
282					0·926	174·5	3·000	53·9			2·140	75·5	1·260	128·1
(293)														
(304)														
316					0·961	195·7					2·265	82·9	1·381	136·3
(329)														
(342)														
355					1·003	218·5					2·735	80·3	1·300	168·5
398					1·030	248·0							1·495	170·5
447					1·130	264·0								
E.P. index number	—		47·7		120·3		67·8		67·1		72·2		128·3	
Min. seizing load, kg	89		112		100		100		141		178		126	

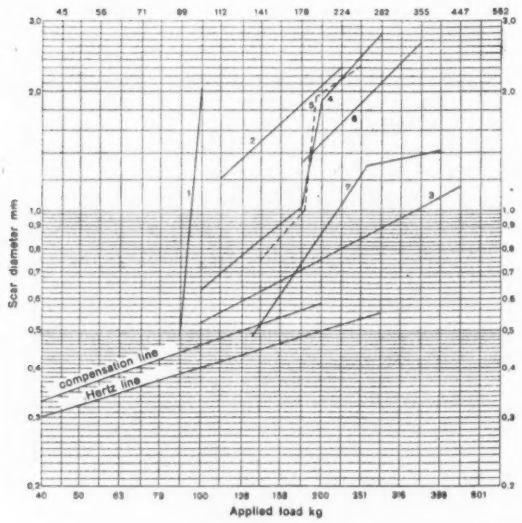

FIG 1

WEAR–LOAD DIAGRAM FOR THE SEVEN OILS TESTED

The lines in this figure are based on the data given in Table IV.
Hertz line drawn according to ref.[1]
Compensation line drawn according to ref.[4]

which each E.P. additive begins to be effective. This temperature is reached at the expense of friction heat generated between the surfaces when seizure occurs.

The temperatures are higher or lower according to that at which the additive comes into action, and linked with the additive's chemical stability. Chemical attack occurs with greater or less ease, even when the additive contains the same elements, according to whether these react more or less readily with the metal surface. The temperature required for the additive to go into action, and the rate of the additive–metal surface reaction, are factors which, together with the applied load, determine the wear-spot diameter on the balls.

The lower the temperature at which the additive starts operating and the quicker the chemical attack, the smaller will be the area of contact on the balls.

In looking at the wear–load diagram it can be seen that the same additive may behave differently according to the level of the applied load, *i.e.* the wear–load curve can reach the highest values in one or more steps (see samples (4) and (5)). At low loads a series of points is obtained corresponding to runs where the

scar remains relatively small and grows only in proportion with the applied load. At a given load value the scar becomes decidedly larger, and the wear–load curve shifts to a higher level zone, here retaining a similar slope to the previous one.

This behaviour might be explained by admitting that the additive acts in a different manner according to the local temperature encountered (flash temperature).

TABLE V

	Copper strip (ASTM)	Iron strip
(3) Base oil + lead naphthenate + 0·5% S . . .	4 *c*	slightly positive
(4) Base oil + chlorinated paraffins	1 *a*	negative
(5) Base oil + additive A .	1 *a*	,,
(6) Base oil + additive B .	1 *a*	,,
(7) Base oil + additive C .	4 *c*	slightly positive

For additives containing more than one active constituent, chemical behaviour will differ with changes in temperature, *i.e.* first one of the active components will go into action, followed by the others one after another.

In the case of sample (3) the curve retains the same slope. This sample contains free sulphur. The ease and rate of attack is such as to form a quite small wear-spot even at very high loads. The type of chemical attack remains unchanged throughout the whole series of tests, and it is evident that the scar diameter is proportional to the applied load. This, however, confirms that the reactivity of a sulphur and chlorine base additive, and hence its good E.P. qualities, are related with corrosivity.

The corrosive properties of the additive oils the authors prepared were determined according to the ASTM method,[6] although this method is valid only for compounds containing sulphur. The tests were performed with both copper and iron strips.

Table V gives the results of this test.

In the samples tested high " E.P. index numbers " were found for those samples which gave a positive result in the corrosion test.

Both sample (3) and sample (7) actually showed a value of 4 *c* on ASTM scale in the corrosion test.

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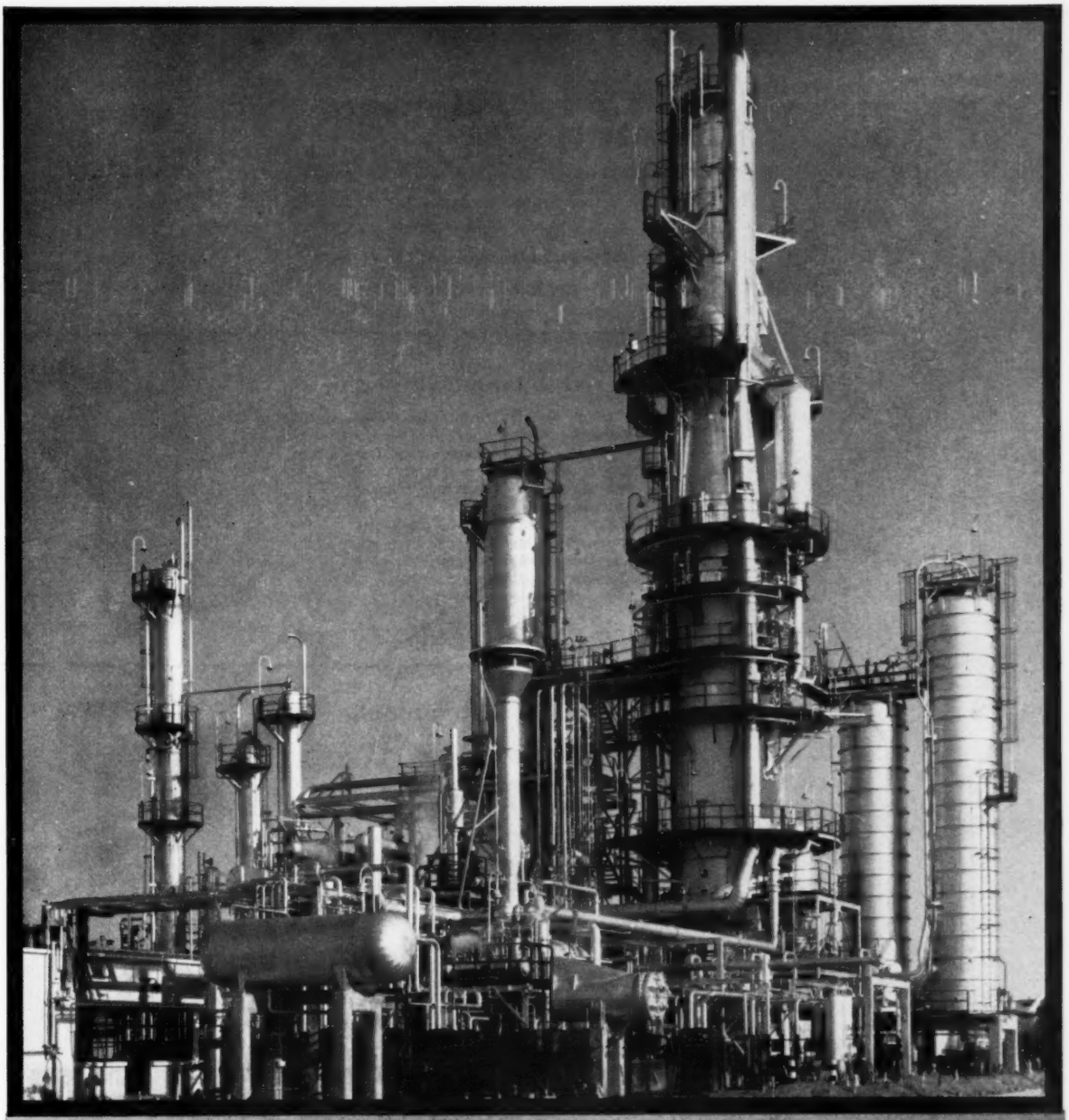

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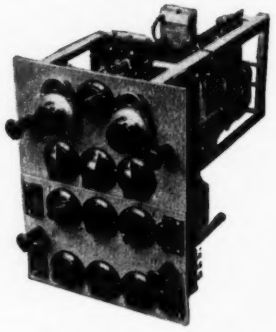

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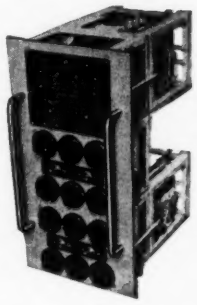

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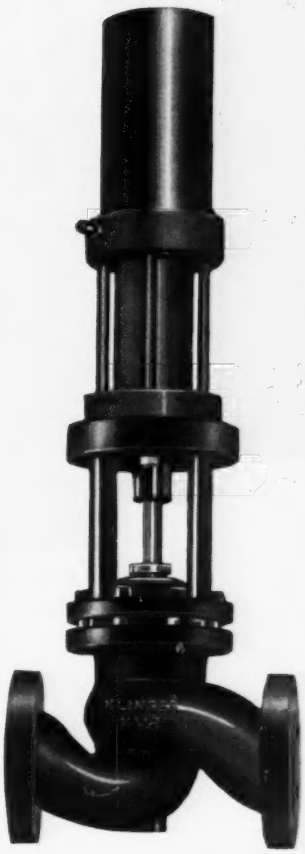

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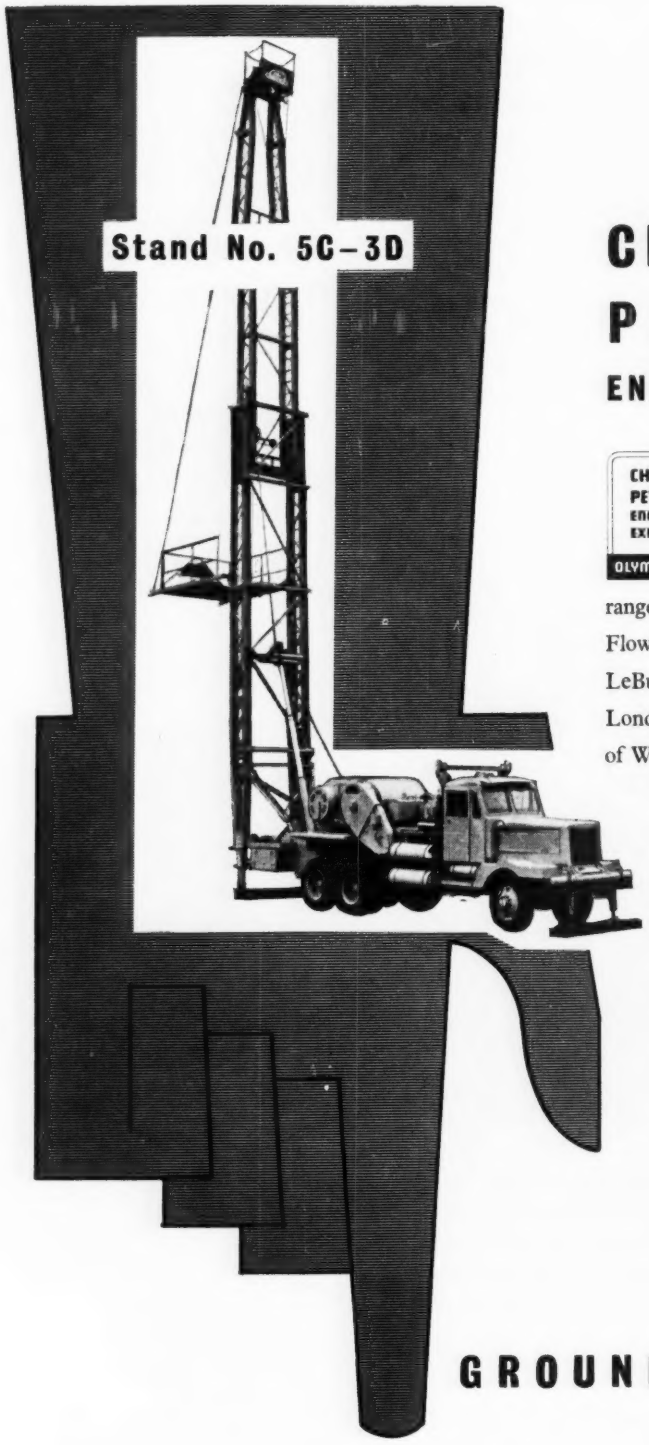

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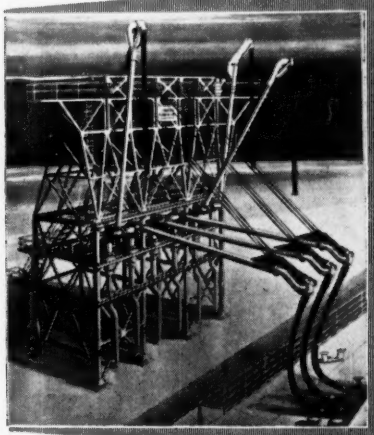

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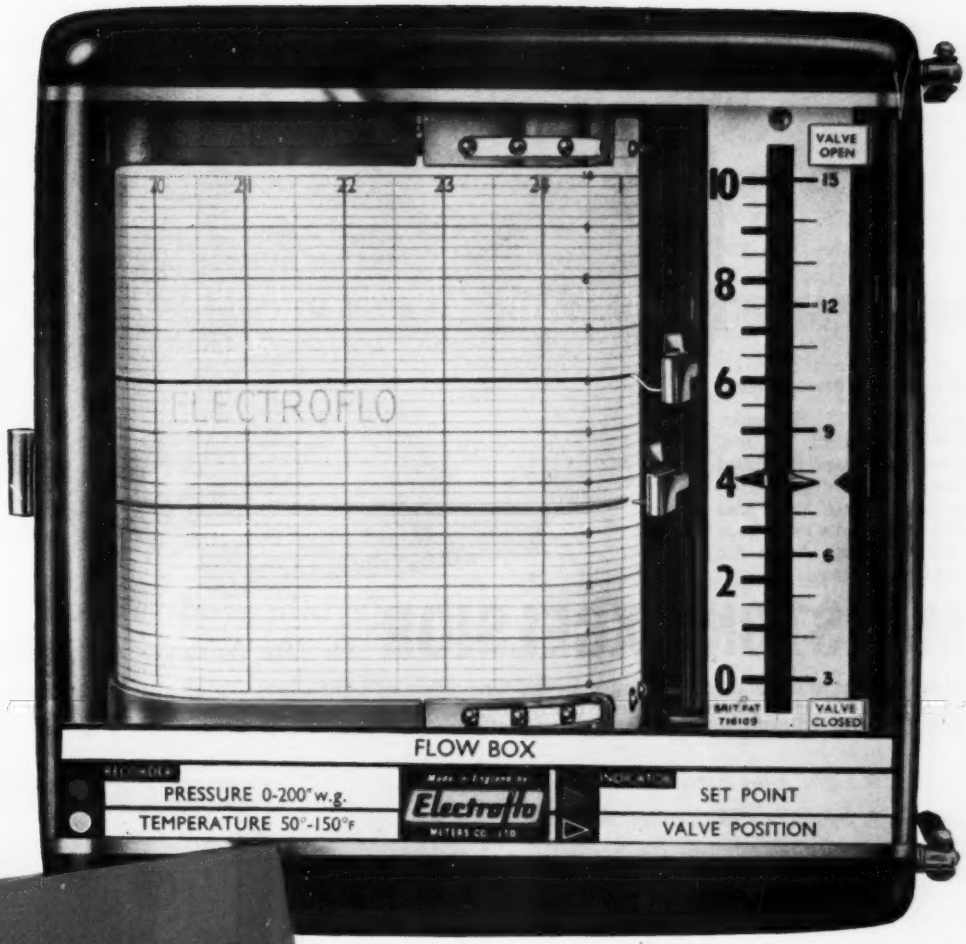

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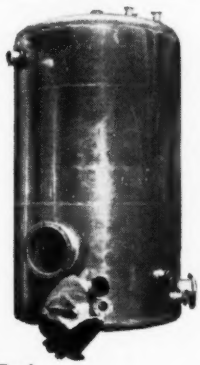

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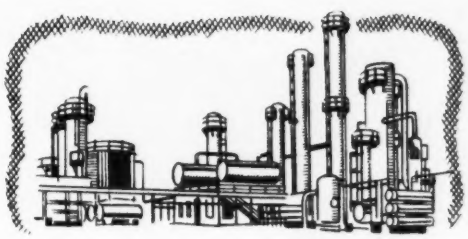

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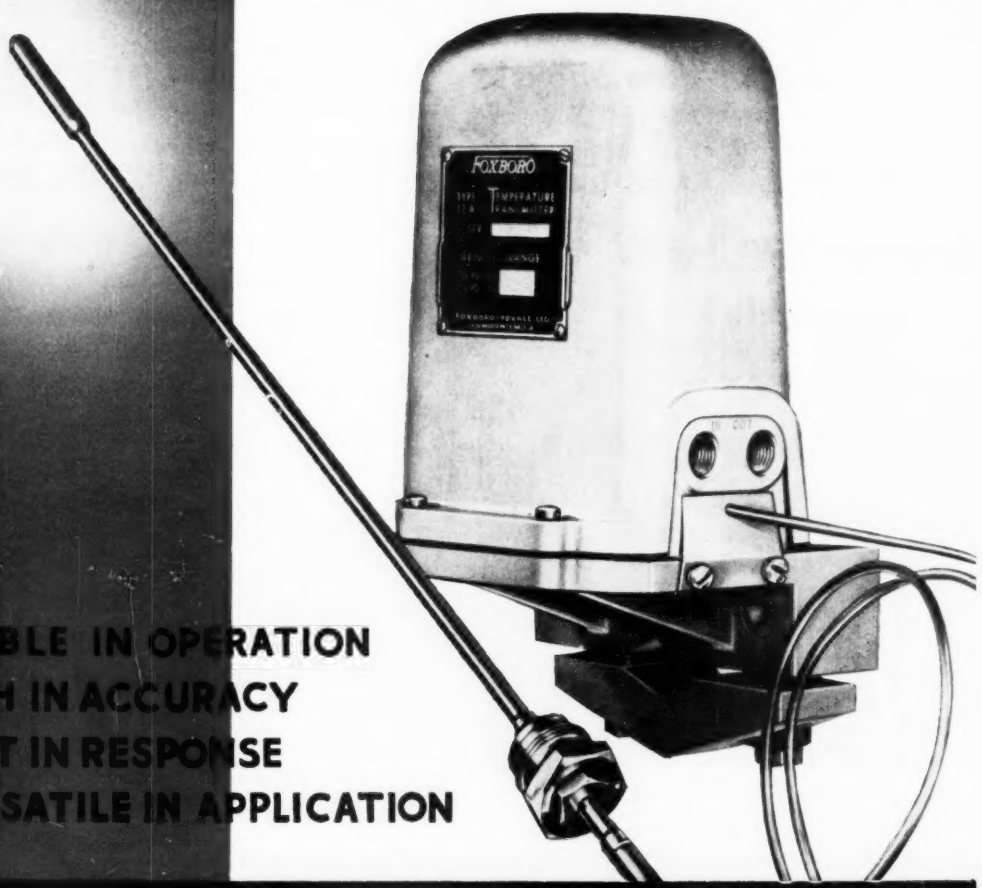

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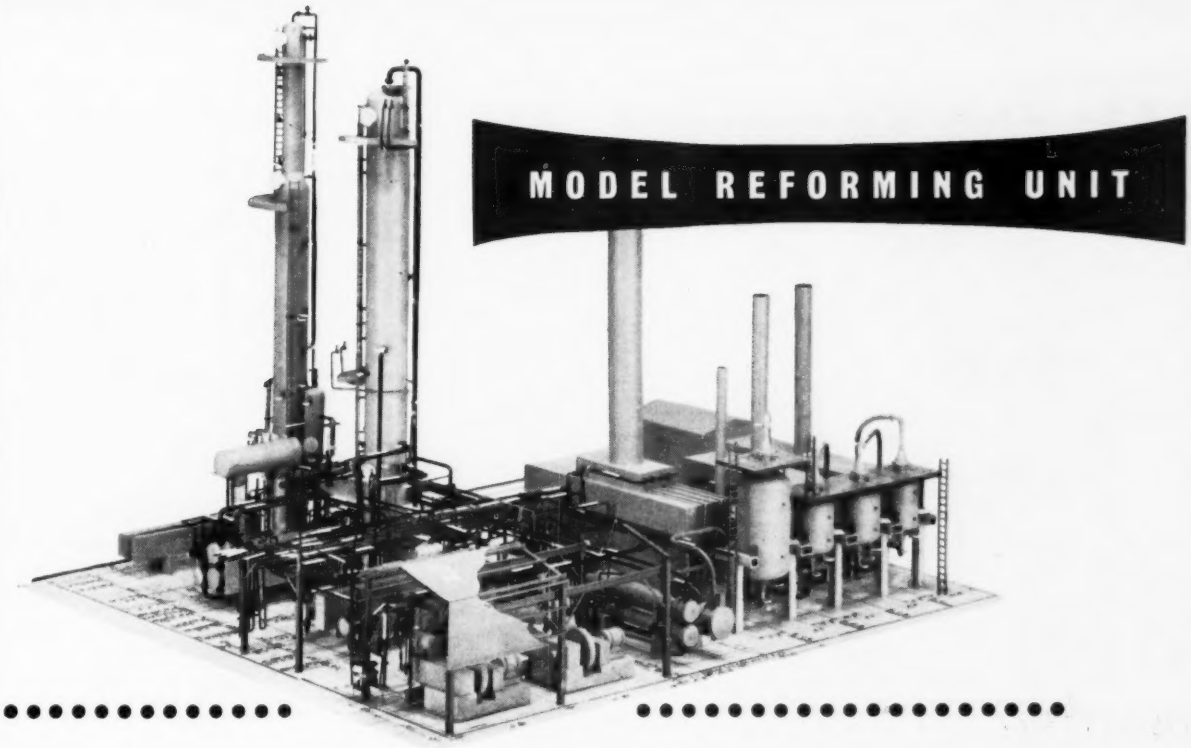

This scale model of a Kellogg catalytic reforming unit designed for Sinclair-Baker catalyst illustrates the detailed thoroughness of Kellogg services. The Kellogg organisation has developed a wealth of experience in engineering and constructing a large number of reforming units of different capacities using as many as seven different catalytic reforming process designs. The most recent Kellogg design for the highest octane reformates utilizes the Sinclair-Baker catalyst.

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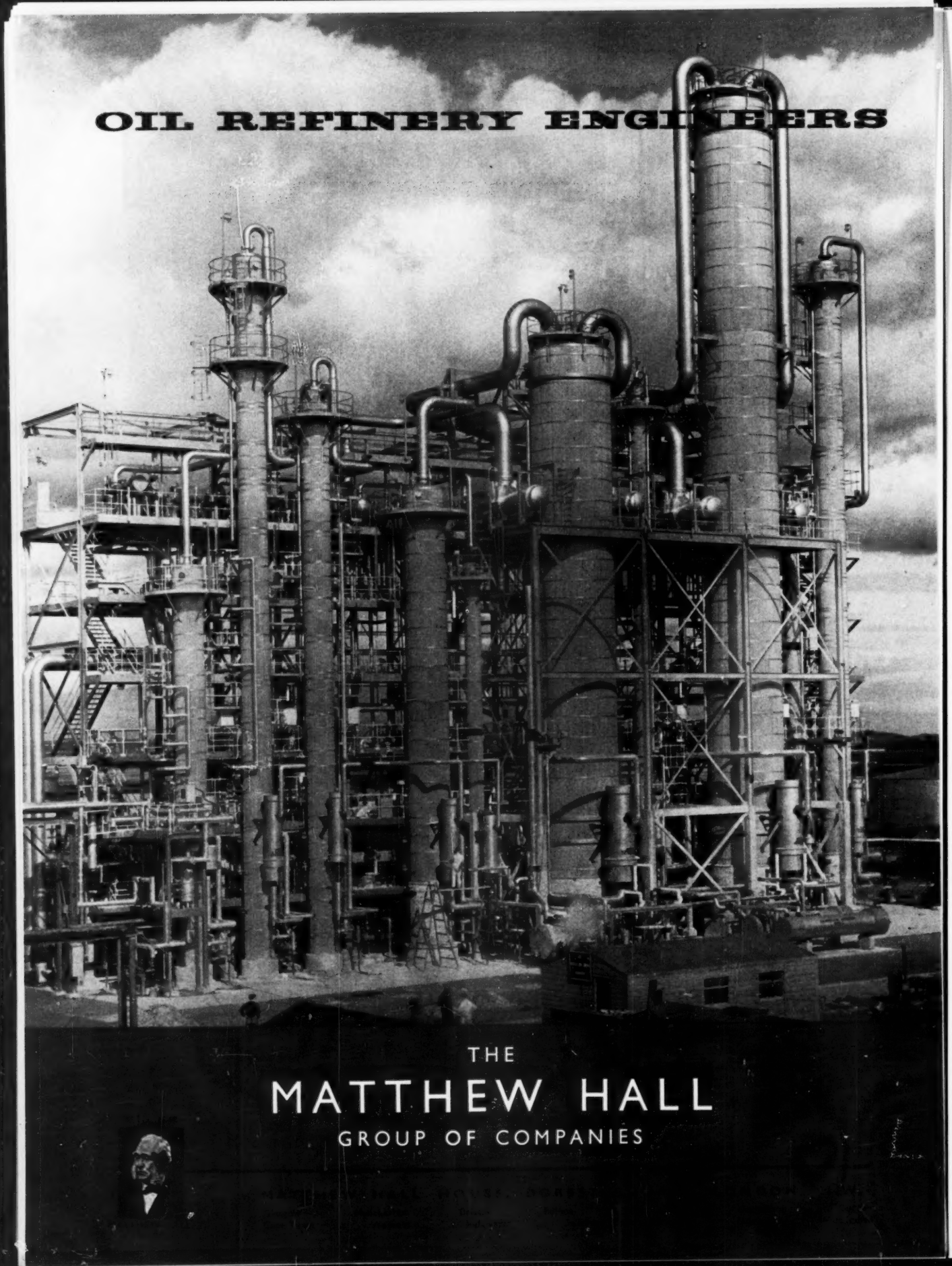
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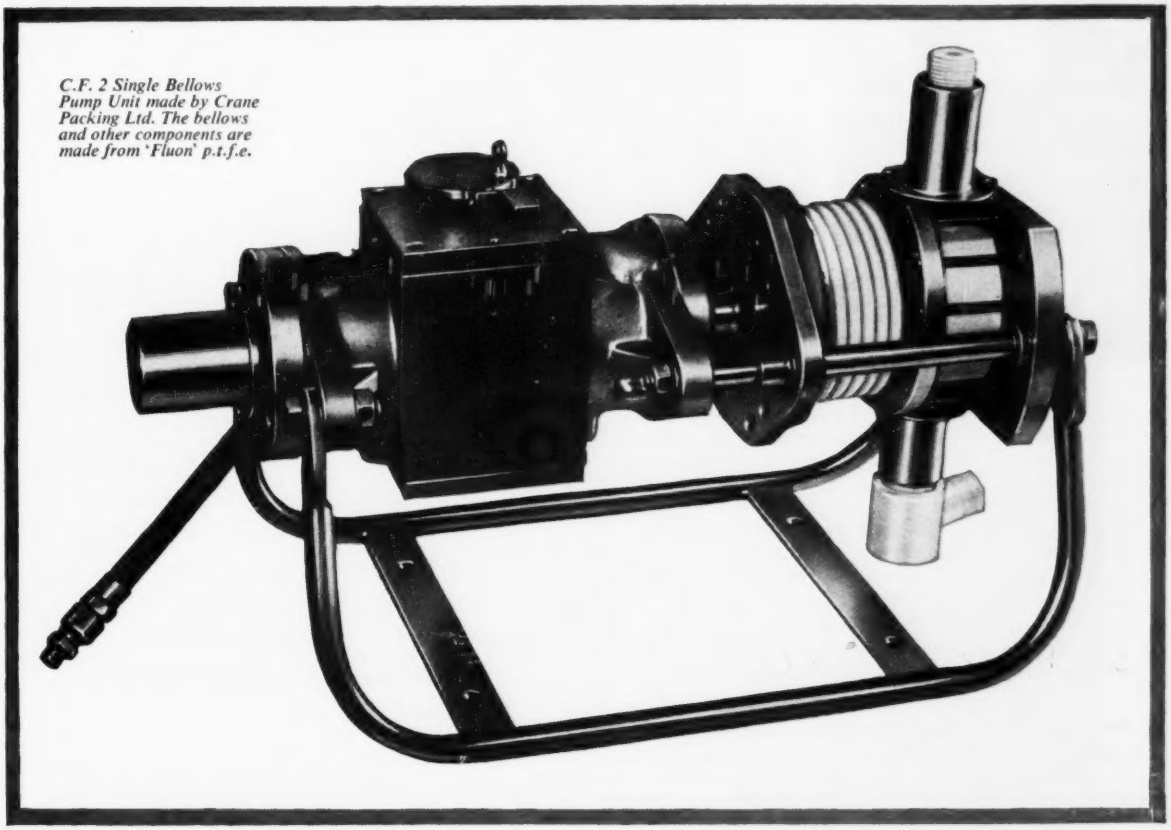

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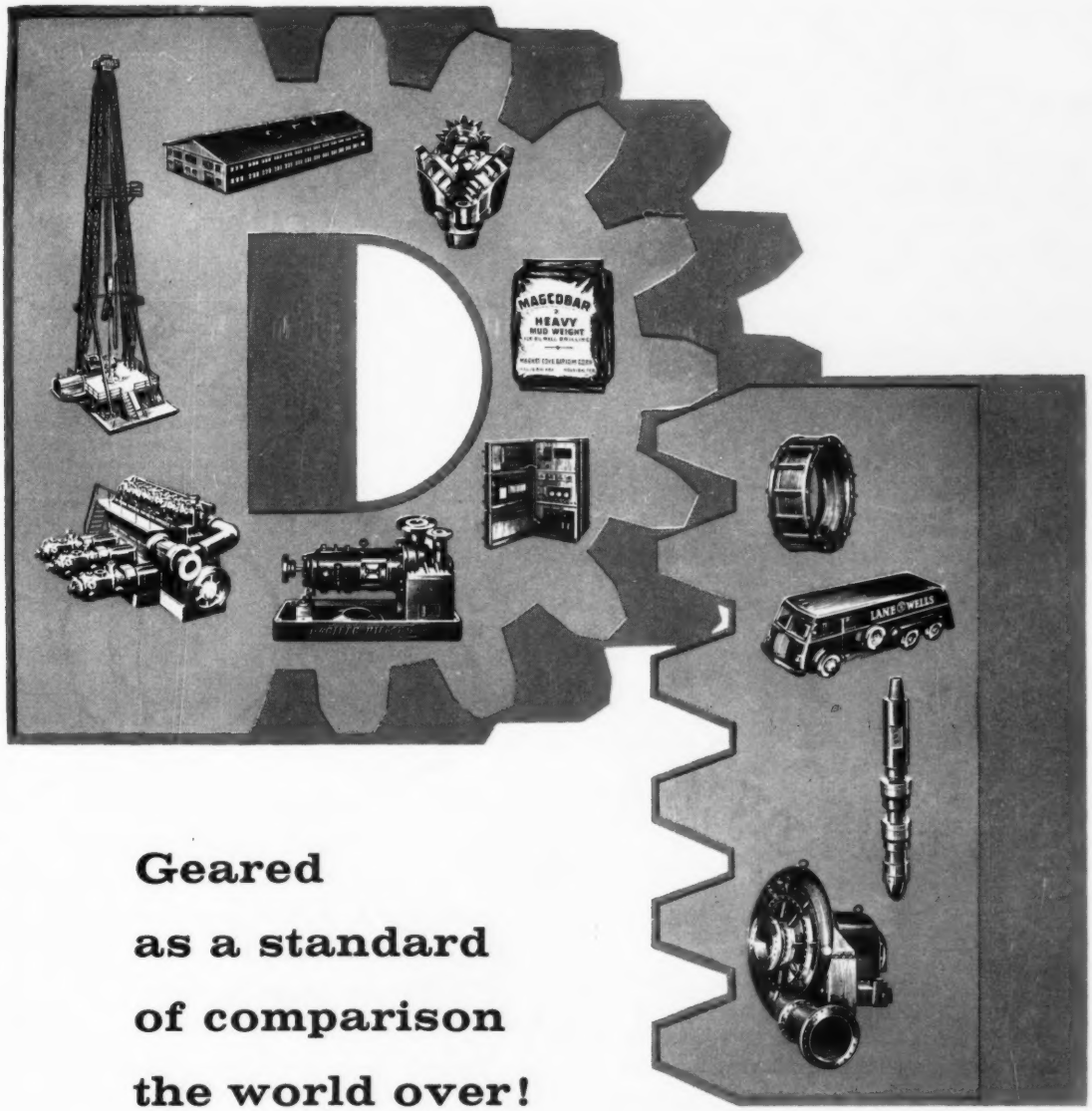

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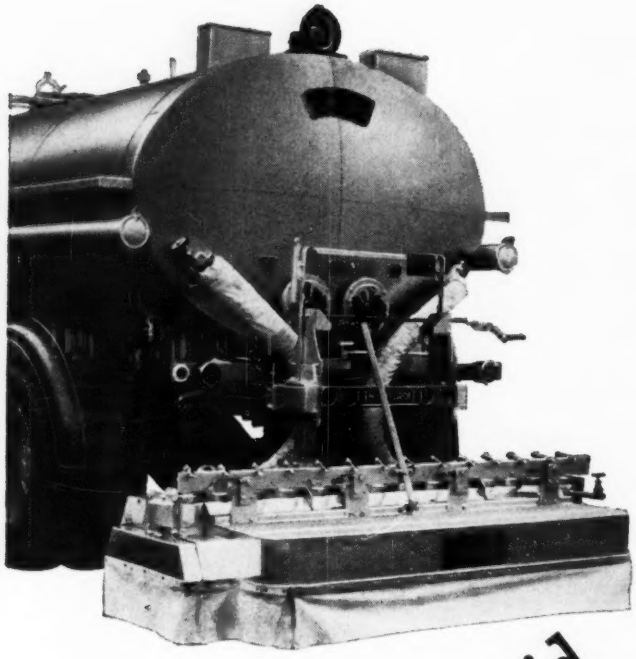
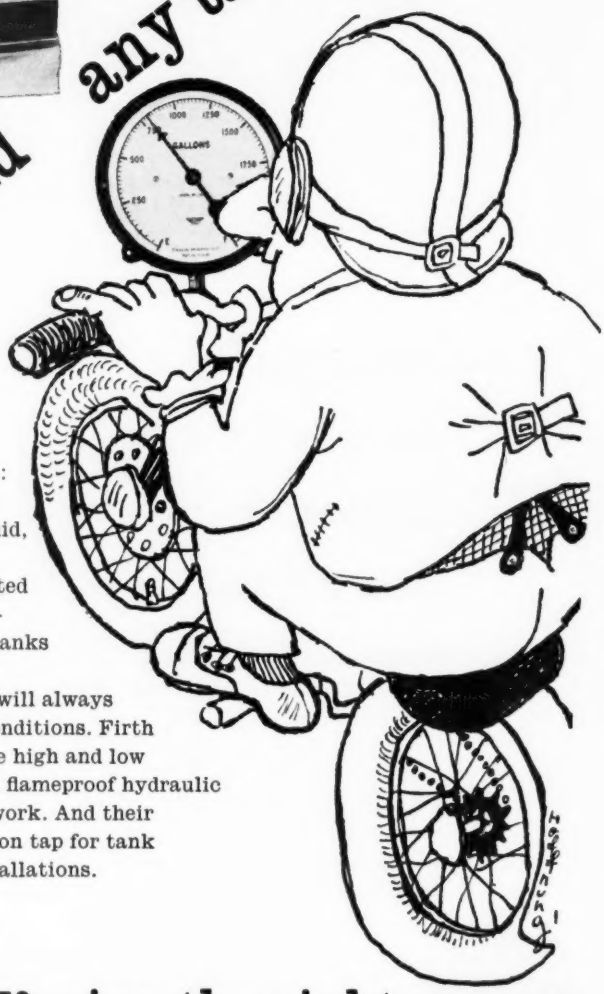


any liquid any tank anywhere



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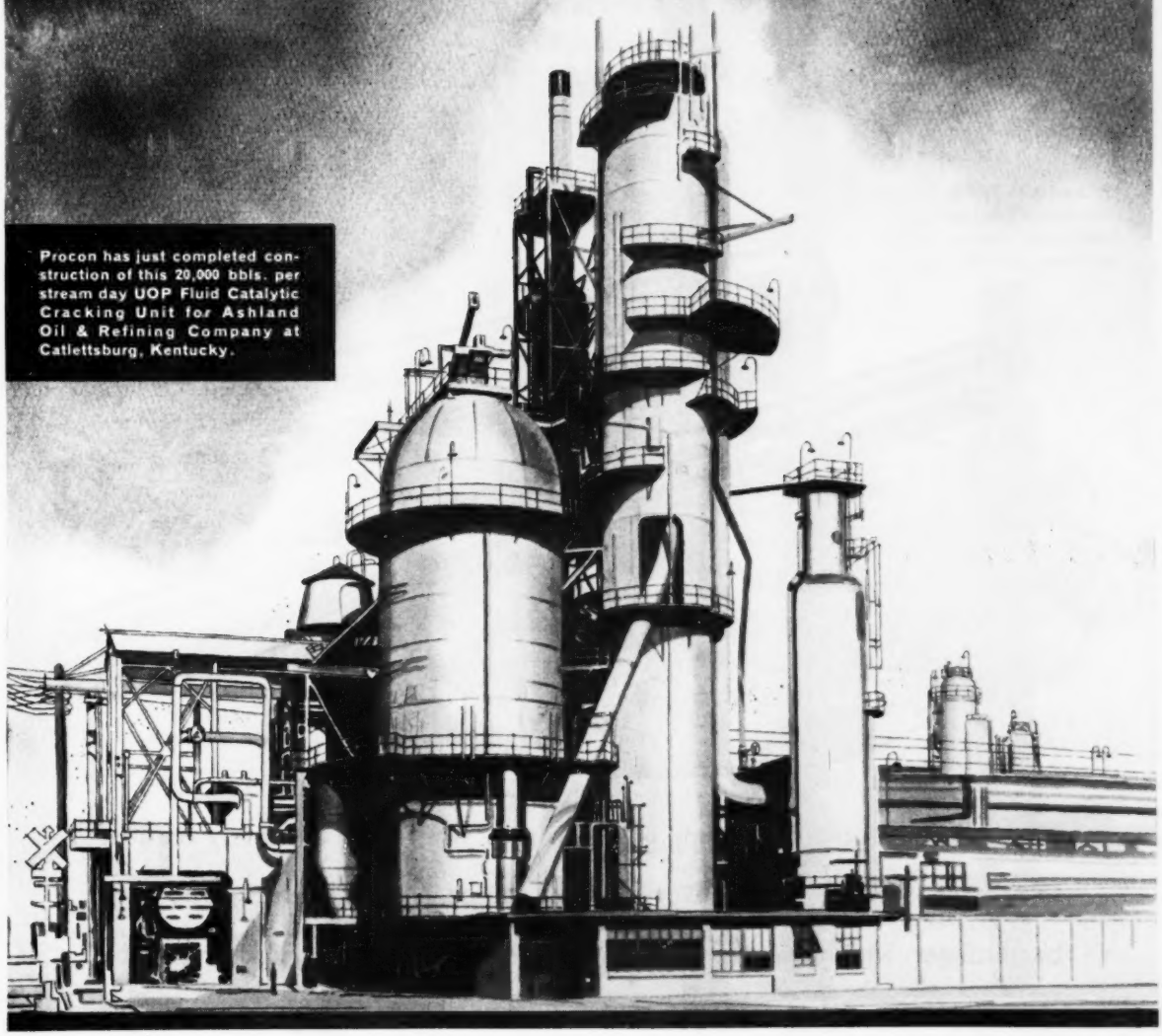

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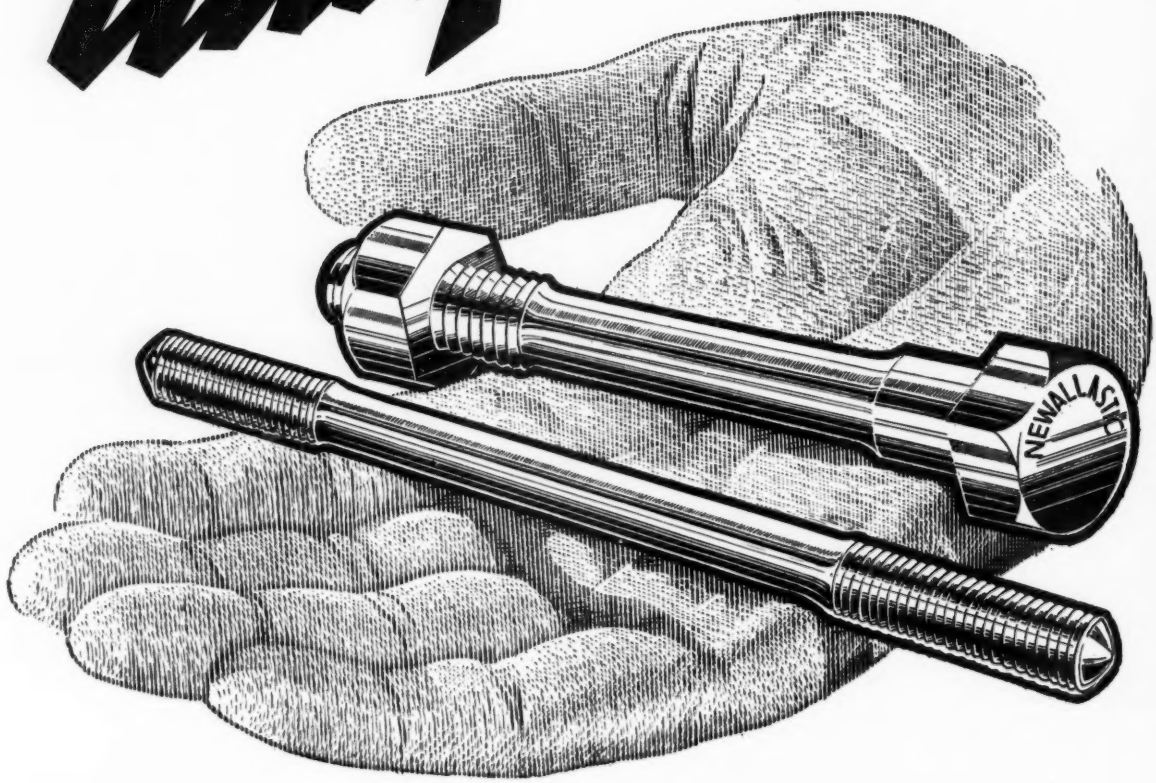

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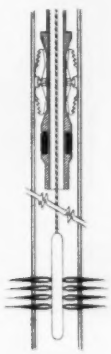

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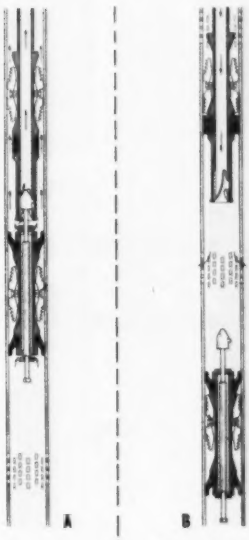
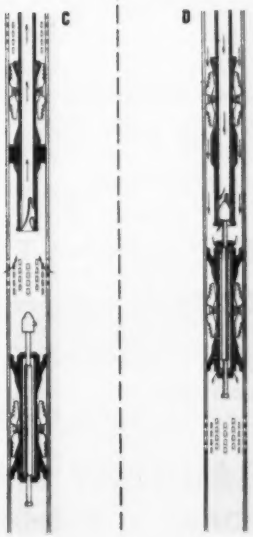


Run tools in together

Release Plug, straddle interval, fracture or acidize

Swab and Test

Pick up plug, move to next interval



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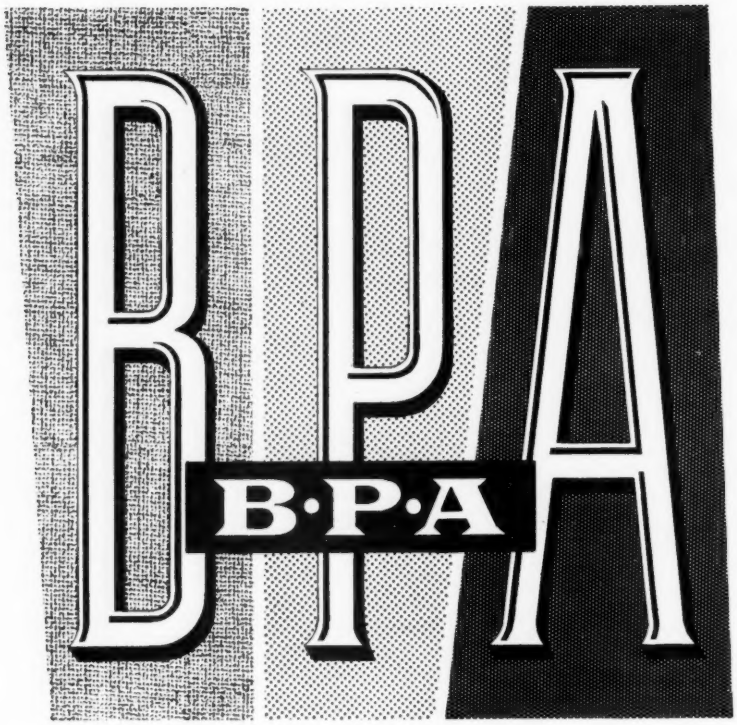

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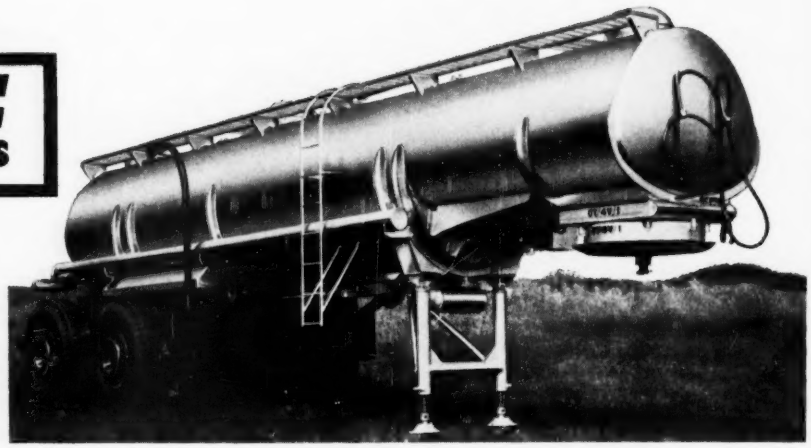

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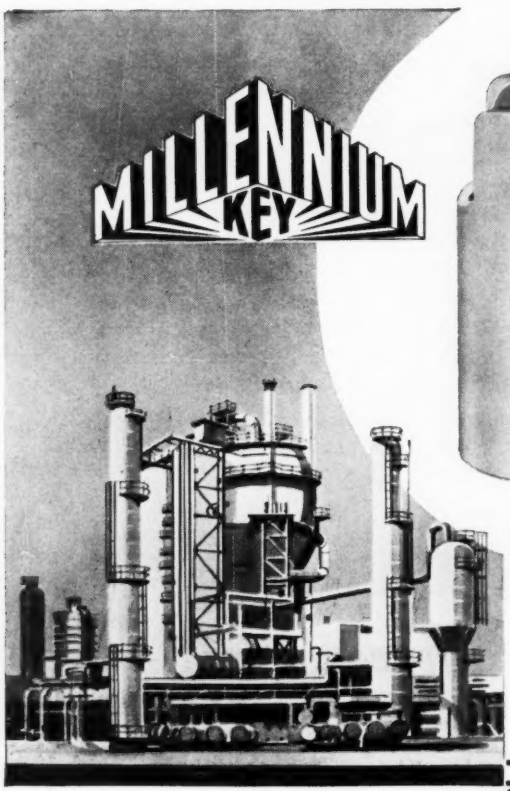

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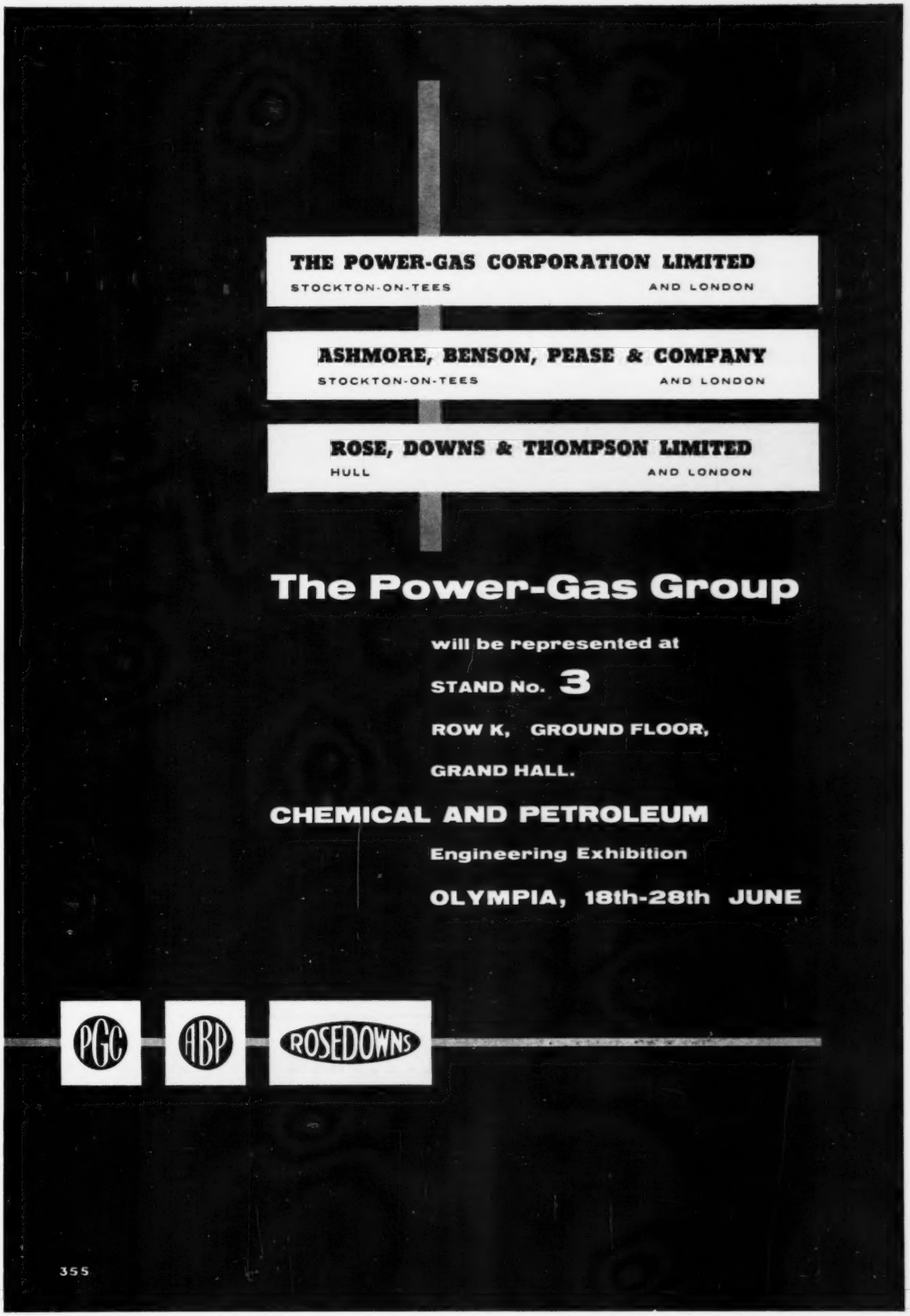
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I'm fed up with filters!
let's sit down...

Let's sit down and talk about cricket, the weather... anything so long as it isn't your confounded Simmonds Fram Separators.

Sorry old man . . . didn't mean to bore you.

That's quite all right. Delighted to know the R.A.F. is getting up to date. So you actually fly your planes on kerosine now instead of water!

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Little strips of erk, for instance?

. . . Anything down to 5 microns.

That gives me a big kick.
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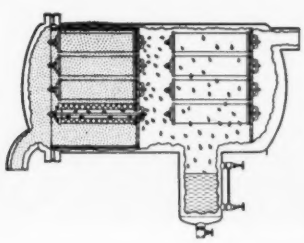

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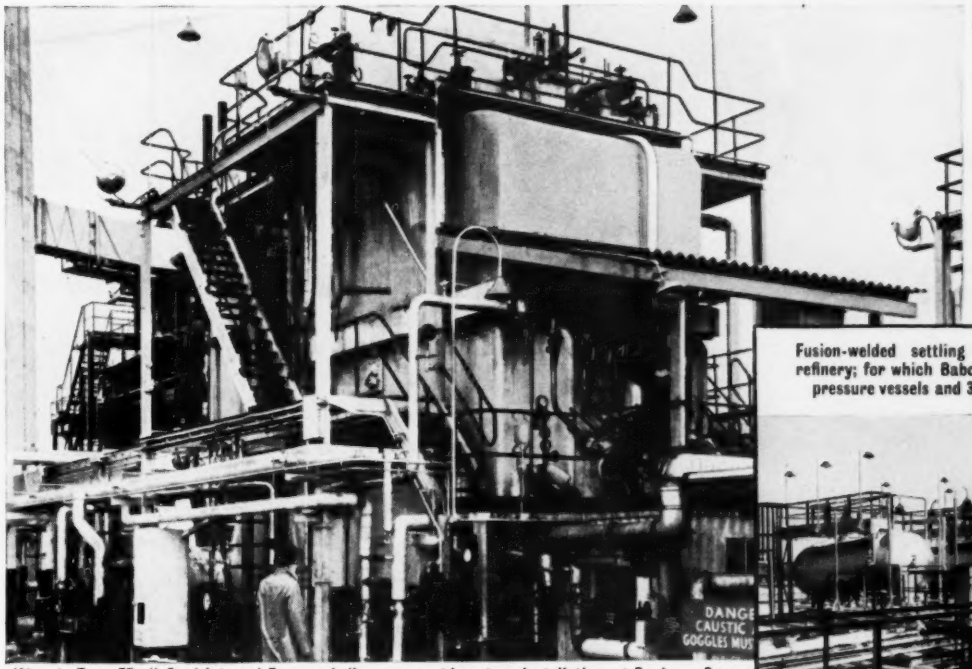
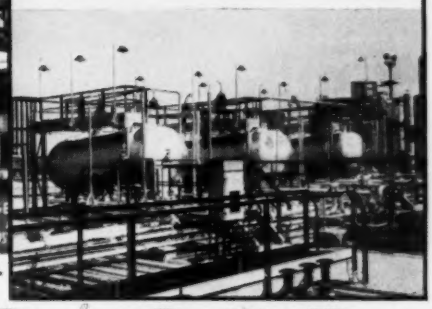


(Above) Type-FP oil-fired Integral Furnace boiler as an outdoor-type installation, at Fawley refinery.



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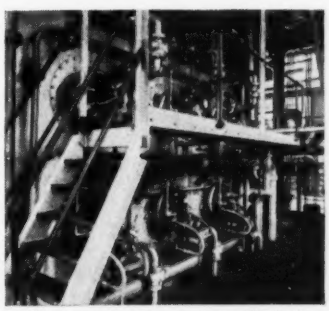
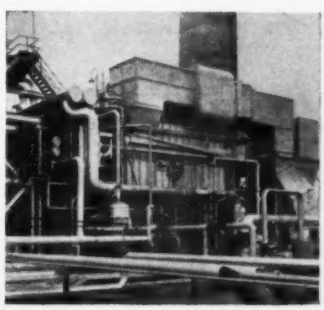
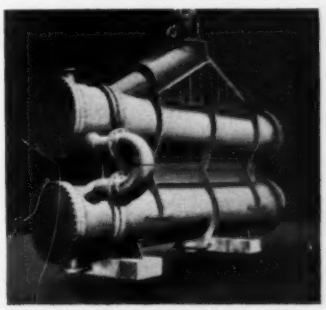


Oil-firing equipment on type FH boiler, Fawley refinery.



Type FH, oil-fired, Integral Furnace boiler as an outdoor installation.



Typical Babcock heat-exchanger for service in an oil-refinery.

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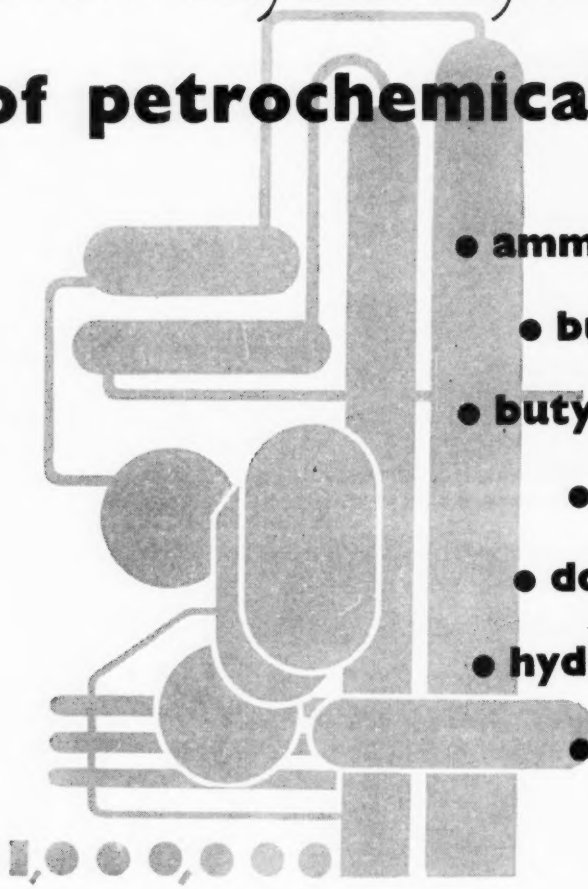

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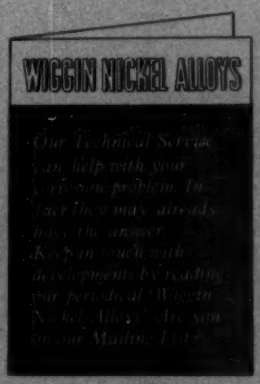

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